TRANSACTIONS

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

VOLUME 42

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TRANSACTIONS

of

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

No. 1025

FORTY-SECOND ANNUAL MEETING, 1936

THE 42nd Annual Meeting of the Society held at the Palmer House, Chicago, Ill., January 27-30, established a record for attendance with a total registration of 1266 members, guests and ladies from all parts of the United States and Canada. Representatives from each of the Society's twenty local chapters were present to participate in the discussions and to attend a meeting of the chapter officers.

Unusually large audiences were present at all of the regular business and technical sessions to discuss the Research Committee Report, the proposed Code for Rating and Testing Air Conditioning Equipment, and the papers presented on a variety of technical subjects. A delightful innovation that was greatly enjoyed was the joint session of the *National Warm Air Heating and Air Conditioning Association* and the Society on January 29.

On January 27 the Council attended a luncheon and meeting at the Saddle and Sirloin Club, Stock Yards Inn, with every member in attendance and then took part in the formal opening of the 4th International Heating and Ventilating Exposition held at the International Amphitheatre. During the five days in which the Exposition was open for inspection, nearly 50,000 people viewed the equipment of over 310 various manufacturers.

Pres. John Howatt, Chicago, called the 42nd Annual Meeting of the Society to order in the Red Lacquer Room of the Palmer House on Monday, January 27, 1936, at 10:00 A.M., and outlined briefly the various important events of the meeting. President Howatt then read his report.

Report of the President

A survey of the activities and affairs of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS for the past year justifies the optimism that is felt by your

officers in looking toward the future. The reports from the different committees of the Society will present a picture of progress and solidification in the year through which we have come that should make all have faith in the foundations and structures that have been built.

The most important asset of this Society is its membership and while the membership is not large when compared with some national organizations, because of the character and interest of the members, it has an effectiveness far beyond that which is accounted for by numbers only. The membership list has been revised until today those whose names appear on it have a real interest in our success.

Our finances are on a sound basis with our assets, so liquid, any emergency can be met. Expenses have been maintained substantially below revenues in order that we may again commence to build our endowment accounts and provide for future security.

The TRANSACTIONS for the years 1933 and 1934 are being printed and within a month will be in the hands of the members. The publication of these TRANSACTIONS is an obligation the Society owes to its members and it is my hope that never again will the publication of these valuable records be delayed as was the publication of the 1933 TRANSACTIONS.

THE GUIDE continues to be of more and more importance in the affairs of the Society. It has become the encyclopedia of technical information for the heating, ventilating and air conditioning engineer and the recognized authority in its field. It is bringing a great deal of favorable attention and comment to the Society as a whole. In addition it has proven to be a source of revenue almost every year, with the sales for the past year exceeding that of any other. The success of THE GUIDE is very largely due to the capable energetic work of our Secretary, A. V. Hutchinson, the work of John James, and the work of the members of the Society who were selected to form the Guide Publication Committee and those who so generously helped this committee in the preparation of text material.

The publication of our papers in the JOURNAL SECTION of *Heating, Piping and Air Conditioning* has continued to demonstrate the wisdom of those who arranged for the original contracts. Dealings with the publishers have been uniformly agreeable and satisfactory since their beginning.

As conditions in the business field continue to improve and our revenues from our publications and other sources increase, the president believes we should, as soon as possible, make arrangements to reduce the dues of members to $15.00 per year. The budget of the Society could even now be prepared on that basis, so I recommend consideration be given to this change in our dues account at an early date. The importance of the work in Chapters is recognized and should be supported and promoted as far as possible. A reduction in the cost of belonging to the Society will make it easier for Chapter groups to obtain working funds from their members.

Our Research program has been maintained as actively as finances permitted. Research is one of the activities that has made this Society outstanding among engineering organizations and should be maintained as extensively as possible. Your President believes a definite policy providing that a certain proportion of the net profits from THE GUIDE be set aside for Research should be established. There should be an extension of co-operative Research agreements with other institutions and a definite policy of conducting fundamental rather than specific Research: Research in the relationship between air environment, physiology, pathology; on the relationship between the environment and human health and well being.

I cannot close this message without expressing my gratitude to the members in conferring upon me the honor of being their Chief Administrative Officer for the past year. Whatever success may have come from my efforts to increase the prestige and influence of the Society, to stimulate the interest in Chapters and bring about a feeling of friendship and cohesion among our geographically wide-spread units has been possible only through the friendly co-operation and support it has been my good fortune to receive. The real work is done by your Council and the regular and Special Committees appointed. It augurs well for the future that we are able to draft members who give freely of their time and talent in the interest of our Society.

JOHN HOWATT, *President.*

The report of the President was recorded and approved as read.

The report of the Treasurer was then introduced and was personally presented by the Treasurer, A. J. Offner, New York.

Report of the Treasurer

Your Treasurer is very happy to report that the treasury is in very good condition. As of December 31, 1935 there was a balance in all funds of over $51,000 and the market value of securities we have amounts to about $20,000, making a total of over $71,000. Rather than quoting a lot of detail figures which you probably would not be interested in and probably would forget as soon as you heard, I would refer those members who are interested in that to the Report of the Finance Committee and the certified public accountant.

A. J. OFFNER, *Treasurer.*

President Howatt ruled that if there was no objection the Treasurer's Report would be made a part of the records and approved as read.

A. V. Hutchinson then read the Report of the Secretary.

Report of the Secretary

In this brief description of the work of the headquarters office it will show that the Society's activities increased tremendously during 1935. A substantial increase in membership was recorded, a larger GUIDE and a greater volume of sales, production of two volumes of TRANSACTIONS, together with normal routine of issuing the monthly JOURNAL, handling memberships, accounts, changes of addresses, codes and other administrative matters has required the careful attention of the office personnel to handle the innumerable details and meet the demands for service. The production of THE GUIDE required a full nine months, the Committee establishing the outline, the reviewing, revising, and editing, before printing, proof reading, binding and distribution. The increased demand for this reference volume had exhausted the entire 1935 edition by November 1 and the Guide Publication Committee will report that the greatest sale of copies ever recorded came during the past year.

An enlarged headquarters office was leased this year and if Society activities continue at the present pace, these quarters will be cramped within another year.

The tremendous public interest in automatic heating and air conditioning has brought a heavy volume of inquiries to the Society's headquarters which has acted as a clearing house in advising of information sources and service.

The work of the Society to obtain employment for members has been actively continued throughout the year with the result that many leads have been given to those who needed employment.

Proposed changes in the Society's Constitution and By-Laws were voted on by members and new rules for Student Member qualifications have been adopted. The result of the voting will be announced by the tellers at this session.

The Roll of Membership compiled for THE GUIDE 1936 records more than the usual number of changes and members are urged to co-operate in keeping the headquarters office informed of changes in location or mailing addresses so that Society mail and publications may reach them promptly.

A net gain in the Society's membership has been recorded and a comparison of the status as of December 31, 1934 and December 31, 1935 is of interest:

	1935	1934
Honorary Membership	1	2
Presidential Members	24	23
Members	1248	1156
Associate Members	407	339
Junior Members	216	233
Student Members	148	97
	2044	1850

Co-operation with Chapters has been stimulated this year and in many cases subjects and speakers have been arranged for by the Society.

Special attention has been given to the newly organized chapters in Washington, D. C., Oklahoma City and Winnipeg, and there is prospect for additional chapters in several other cities where local members feel that interest has reached the point where a Society organization can function effectively.

In accordance with custom, the headquarters staff in co-operation with the Program Committee and the Local Chapter Committees worked out details of the Annual and Semi-Annual Meetings.

It is with genuine pleasure that we acknowledge the co-operation given by the Officers, Council members, Chapter Officers and Committees, as well as the Research Laboratory who have made 1935 a successful year for the Society.

Acknowledgment should also be made for the assistance of the headquarters office for their interest, loyalty and faithful performance of manifold duties required to successfully handle the administration activities of the Society.

Respectfully submitted,

A. V. HUTCHINSON, *Secretary.*

Mr. Howatt ruled that if there were no comments or objections the report would be published as read.

The Secretary then presented the Report of the Council.

Report of Council

Since the last Annual Meeting, three meetings of the Council have been held to carry on the business of the Society and the membership will be glad to know that definite progress was made towards strengthening the Society's membership and its financial condition.

At the organization meeting of the Council held in Buffalo, President Howatt announced committee appointments in accordance with the provisions of the Constitution and By-Laws, the appointment of A. V. Hutchinson, Secretary, was confirmed, depositories for Society Funds were selected and the Budget for 1935 approved by the Finance Committee was adopted. The Budget estimate indicated total income of $59,633.24 and an estimated expenditure of $58,650.00. The report of the Finance Committee will indicate that a greater revenue was produced and expenditures were proportionately maintained.

The Council appointed the F. Paul Anderson Award Committee in 1935 and the unanimous choice of the Committee reported to the Council and approved was that Dr. Arthur Cutts Willard, Urbana, Ill., president of the University of Illinois receive the F. Paul Anderson Medal for 1935. The presentation was to be made by President Howatt at the 42nd Annual Banquet, January 29, 1936.

One of the most important actions of the Council was the co-operative movement started by the Council to encourage greater interest in Society activities and the means was provided for an official delegate from each Chapter to be present at this meeting.

Another significant action was the authorization of an exchange service plan to members in co-operation with the *A. S. R. E.* for the year 1936 and, since the announcement was sent out in December, 200 members have indicated their desire to participate in this plan.

To extend the scope of the Society's technical activities, John W. James joined headquarters office staff in April.

At the June Meeting of the Society, a detailed report was given on the progress of the Fourth International Heating and Ventilating Exposition and the Society is indebted to the members of the Advisory Exposition Committee for the splendid co-operation that they rendered to the management of the Exposition, in order to produce the great spectacle which opens today at the International Amphitheatre.

As required by the Society's Constitution and By-Laws the Council established the dues rate for the year 1936 and announced nominations for five members of the Committee on Research to serve for a three-year period commencing in 1936.

The invitation of the Philadelphia Chapter to hold the Semi-Annual Meeting 1936

at Buck Hill Falls, Pa., was approved by the Council and announcement has been made that this will be a joint meeting with the *A. S. R. E.* on June 22 to 24.

Due to improved financial conditions some progress was made in changing Society investments in accordance with recommendations from the Finance Committee, so that practically all surplus funds of the Society are now in U. S. Government securities as required by the By-Laws.

During 1935 three new chapters were successfully launched; the Council granting Charters for local chapters to members in Washington, D. C., Oklahoma City and Winnipeg.

As required by the Constitution and By-Laws the Council confirmed the election of members in accordance with recommendations of the Committee on Admission and Advancement. For the twelve months of 1935, 358 applications were received. It was also necessary for the Council to act on the resignations of 45 members during the year and to cancel 73 memberships in accordance with the By-Laws.

The Council regrets the death of its first President, Stewart A. Jellett, and several of its oldest members who joined the Society over 25 years ago: C. W. Christian, R. Collamore, G. M. Getschow and J. D. Small.

The Society has passed through a year of substantial growth and prosperity and the Council wishes to acknowledge the interest and cooperation of the various Committees and individual members who have contributed to its progress in 1935.

Respectfully submitted,

THE COUNCIL.

There being no changes or modifications to the report, President Howatt said that it would be published as read.

Report of the Finance Committee was introduced by E. H. Gurney, Toronto.

Report of the Finance Committee

The accounts of the Society for the year 1935 reflect not only the general improvement in business conditions, but also the great increase and the particular interest in the field of engineering covered by the Society.

Your Society shows a surplus for its year's activities of approximately $10,000, which is, however, reduced by two special appropriations: one of $600 for editorial services, and the other of $1500 for the Research Fund. It is recommended that this surplus be used for the purpose of restoring the Endowment Fund to a figure close to its original proportions.

After a year of delightful co-operation with your President, I find that I am not quite at one with him in respect to this matter of fees. True, we had this surplus last year. It is however equally true that the budget, which will be presented by this Committee to your incoming Council, will show no surplus. The growing field of activity to be covered by the Society, plus the fact that last year everybody in the Society's employ had to work at almost too high tension, means that, if we do our duty, we cannot be sure of looking forward to any surplus and immediate reduction in dues and, therefore, in my opinion we must not jump to the conclusion that dues must be reduced immediately.

The improvement in revenue for the Society comes largely from two sources. One, the collection of dues from membership which is up from $15,870 in 1934 to $19,438 in 1935. The other source is from the publications. Here, there is another factor which has worked for improvement in addition to the conditions referred to above: namely, in the improvement in THE GUIDE itself. This revenue is up from $32,371 in 1934 to $47,309 in 1935.

In expenditures there are no important modifications that should be commented on, other than that occasioned by the increased volume of the publication output.

Your Committee has seen fit to modify the endowment investments of the Society by meeting the new By-Law requirement for government securities in liquidating certain bonds and reinvesting. We now have left only one non-trustee security, which your Committee recommends that the incoming Finance Committee shall also convert to government bonds.

There is only one phase of the financial picture which still shows a discouraging

result: namely, the contributions from private individuals or companies to Research, either earmarked for a specific purpose or for general research. Virtually, something over 90% of all contributions received for this purpose last year came from one organization and for a particular project. Having regard to our past record along these lines, it would appear that many organizations, affiliated with the A. S. H. V. E. do not realize or appreciate the value of the service which might be rendered them through the Research Laboratory and co-operating institutions.

Your Committee desires to summarize the balance sheet and income and expense statement from the certified copy of the accounts audited by Price, Waterhouse and Co., which will be filed as a matter of record with the Secretary of the Society.

There have been numerous modifications in the Society's office methods in the last year inherent to a rapidly growing business and the writer would like to record his appreciation of the way the staff has taken the extra thrust, including very heavy demands on their time.

All of which is respectfully submitted,

E. H. GURNEY, *Chairman,*
THE FINANCE COMMITTEE.

Mr. Gurney then moved the adoption of the report and it was regularly seconded and carried and made a matter of record.

BALANCE SHEET* EXHIBIT I

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

December 31, 1935

ASSETS

GENERAL FUND ASSETS			
Cash in Banks and on Hand		$38,441.46	
Cash Deposit in Closed Bank		500.00	
Interest Accrued on Investments		61.42	
Investments in Marketable Bonds at Cost (Market Value at December 31, 1935, Approximates $11,600.00)		11,525.31	
Members' Dues Receivable	$17,936.37		
Deduct—Reserve thereagainst	13,500.00	4,436.37	
Owing by Advertisers and Sundry Debtors	$28,017.53		
Deduct—Reserve thereagainst	2,500.00	25,517.53	
Inventories of Publications and Emblems, at Cost or Lower		2,750.39	
Furniture and Fixtures	$ 5,433.95		
Deduct—Reserve for Depreciation	4,314.42	1,119.53	
Library, at Nominal Value		300.00	
Prepaid Expenses		518.99	
			$85,171.00
ENDOWMENT FUND ASSETS			
Cash in Bank		$ 5,248.14	
Interest Accrued on Investments		143.00	
Investments in Marketable Bonds, at Cost (Market Value at December 31, 1935 Approximates $8,400.00)		11,918.65	
			$17,309.79
THE F. PAUL ANDERSON MEDAL FUND ASSETS			
Cash in Bank		$ 1,086.40	
Cash Deposit in Closed Bank		33.22	
			$1,119.62

* Accompanying report of Price, Waterhouse & Co., dated January 21, 1936.

ASSETS (*Continued*)

RESEARCH FUNDS ASSETS			
ENDOWMENT FUND ASSETS			
Cash in Bank		$ 443.06	
Cash Deposit in Closed Bank		213.53	
			$656.59
UNRESTRICTED FUND ASSETS			
Cash in Bank and on Hand		$ 5,815.10	
Owing from General Fund on Dues Collected, etc		2,363.52	
			$8,178.62
			$112,435.62

LIABILITIES

GENERAL FUND			
Accounts Payable		$ 6,817.83	
Prepaid Dues		1,624.06	
OWING TO RESEARCH FUND			
On Dues Collected	$ 2,363.52		
On Dues when Collected	1,972.52	4,336.04	
Prior Years' Dues of Reinstated Members, to be Collected		307.00	
Reserves for Publications		22,400.00	
FUND BALANCES			
At January 1, 1935	$41,490.53		
Add—Excess of Income over Expenses and Appropriations for the Year Ending December 31, 1935	8,195.54	49,686.07	
			$85,171.00
ENDOWMENT FUND			
FUND BALANCE AT JANUARY 1, 1935			
Principal	$16,686.13		
Deduct—Loss on Sale of Investments	322.85	$16,363.28	
Unexpended Income	$ 980.84		
Add—Interest on Investments	628.17		
	$ 1,609.01		
Deduct—Appropriation to Research Fund	662.50	946.51	
			$17,309.79
THE F. PAUL ANDERSON MEDAL FUND			
FUND BALANCE AT JANUARY 1, 1935			
Principal		$ 1,000.00	
Unexpended Income	$ 96.01		
Add—Interest on Bank Deposit	23.61	119.62	
			$1,119.62
RESEARCH FUNDS			
ENDOWMENT FUND			
Principal		$ 600.00	
Unexpended Income, January 1, 1935	$ 46.94		
Add—Interest on Bank Deposit	9.65	56.59	
			$656.59
UNRESTRICTED FUND			
Balance, January 1, 1935		$ 7,484.18	
Excess of Income over Expenses for the Year Ending December 31, 1935		694.44	
			$8,178.62
			$112,435.62

EXHIBIT II

STATEMENT OF INCOME AND EXPENSES AND APPROPRIATIONS

Year Ending December 31, 1935*

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

GENERAL FUND

INCOME		
Current Year Dues		$30,649.92
DEDUCTIONS AS PROVIDED IN BY-LAWS		
Members' Subscriptions to JOURNAL	$ 4,146.72	
Contributions to Research Fund—40% of Dues of Members and Associate Members Only	11,145.28	15,292.00
		$15,357.92
Prior Years' Dues Collected from Reinstated Members		144.00
Editorial Services		12,800.04
Interest on Investments and Bank Deposits	$ 985.94	
Less—Loss on Sale of Investments	931.69	54.25
Office Furniture—Profit on Sale		71.50
PUBLICATIONS		
Sales (net) of GUIDE—1935 edition	$21,941.72	
GUIDE Advertisements—1936 Edition	25,367.93	
Sales of TRANSACTIONS	156.20	
Sales of Reprints, Books and Pamphlets	250.39	47,716.24
Miscellaneous Income		99.37
Total Income		$76,243.32
EXPENSES		
PUBLICATIONS		
GUIDE 1936 Edition—Printing, Binding and Distribution, including Provision of $12,500.00 therefor	$26,758.59	
GUIDE 1935 Edition—Distribution, etc	4,667.54	
TRANSACTIONS 1933 and 1934 Editions—Additional Provision for Publication	966.02	
TRANSACTIONS 1935 Edition—Provisions for Publication	3,000.00	
YEAR BOOK—Printing, etc. including Provision of $700.00 therefor	1,021.61	
Codes—Printing, etc	57.28	$36,471.04
Salaries	$13,194.00	
Rent and Light	2,583.37	
Postage	1,941.92	
Meetings	834.60	
Chapter Meeting Allowance	1,000.00	
President's Fund	1,063.39	
Traveling—Secretary	951.89	
General Printing	400.00	
Multigraphing	374.67	
Telephone and Telegraph	791.31	
Office Supplies	417.43	
General Office Expense	555.01	
Awards	240.21	
Discounts	353.31	
Professional Fees	500.00	
Depreciation of Furniture and Fixtures	534.97	
Provision for Doubtful Accounts	3,490.55	
Miscellaneous	250.11	29,476.74
Total Expenses		$65,947.78

* Accompanying report of Price, Waterhouse & Co., dated January 21, 1936.

Excess of Income over Expenses		$10,295.54
SPECIAL APPROPRIATIONS AUTHORIZED BY COUNCIL		
For Editorial Services	$ 600.00	
For Research Fund	1,500.00	2,160.00
Excess for Year of Income over Expenses and Appropriations—General Fund		$ 8,195.54

The Membership Committee Report was presented by F. C. McIntosh, Pittsburgh, Pa.

Report of Membership Committee

Probably due to the change in the times, the work of this Committee has been unusually easy and agreeable during the past year. Following the idea of President Howatt, and the general thought that a Society like ours should not expand too rapidly, we have conducted no membership campaign. We have, however, developed and maintained a card file of men who might be eligible. This now consists of over 1700 names. When we learned from any source that a man was in the field of heating or air conditioning, a card was prepared for him and put in the file, where it remained until an application was received, or until we had definite reason to believe he was not interested.

Each of these men has received at least one letter from us. In most cases, the first letter was one briefly mentioning the Society and its purpose, with a request for reply as to whether or not the recipient was interested in receiving further information. If the answer was affirmative, a second letter was sent out with full details of the advantages and obligations of membership. If this letter was not answered, a follow-up was sent to ask if any more information was desired. As an indication of the value of this system, we report that about 40 per cent of the members elected during the year had been listed.

Fifty per cent of those who were sent our initial letter, replied—a very good response. Of these replies, three per cent indicated definite lack of interest, and their cards were removed. Some reported an intention to apply for membership later, but an inability to do so at the time, for financial or other reasons. The file card was marked accordingly.

Members elected to the various grades during the calendar year 1935 are as follows:

Members	127
Associates	79
Juniors	55
Students	91
Total	352

By the twentieth of this month, the Council was voting on approximately forty more.

Although the general growth for the year was healthy, the outstanding membership event was the receipt of applications for four new Chapter Charters. Three of these have been granted; one for Manitoba, Canada, one for Oklahoma City, Okla., and one for Washington, D. C., and the fourth, from Baltimore, is under consideration. There is a great deal of work to be done by a local group in forming a chapter and it is often largely assumed by a few men. The Society is particularly indebted to Colonel W. A. Danielson for the large District of Columbia Chapter; to Professor E. F. Dawson, and to F. X. Loeffler for the Oklahoma group, to C. H. Turland for the second Canadian Chapter, and to J. E. Seiter for the proposed application from Baltimore.

It is our belief that our Society's healthy growth in size and in influence, is best promoted by formation of additional chapters, and we hope that in the near future

this action can be taken in such districts as Atlanta, Dallas, Denver, Indianapolis, Montreal, New Orleans, Omaha and San Francisco.

In considering growth, this committee has always tried to keep in mind the fact that mere increase in numbers leaves much to be desired. Every man elected who is more valuable to us than our present average member, raises the standing by a certain amount, and just as truly, every man elected who is below this average, lowers it. It is quite possible that the best result would be obtained by limiting our membership to a certain maximum number, for a determined period. Such action would discourage the promiscuous solicitation of members and encourage the selection of the best men.

From the viewpoint of our membership work, we respectfully submit the following recommendations to our successors and to the Society as a whole: We recommend:

1. That additional chapters be formed just as rapidly as possible, wherever and whenever the membership is sufficient.

2. That the National Organization take a greater interest in the Chapters, especially the younger ones, and that this interest be expressed through a special committee, similar to the former Chapter Relations Committee. The re-establishment of the meeting of the Chapter Representatives, and the efforts made to promote the attendance of Chapter Officers at this Annual Meeting are steps in the right direction, but something more should be done.

3. That the Chapters, in their regular sessions and their committee meetings, give more consideration to the affairs of the National Organization. One result might be the better preparation of the Nominating Committee members.

4. That for the reasons mentioned in the body of this report, a limit be placed on the number of Student Members and another limit on the total of other members elected, regardless of grade, and that the time and figures for these limits be set by the Council.

The reader takes this occasion to remark parenthetically, and gratefully, that this has been by no means a one man committee.

F. C. McINTOSH, *Chairman,*
JOHN CASSELL,
W. A. RUSSELL,

MEMBERSHIP COMMITTEE.

President Howatt then briefly commented on the Report of the Membership Committee and indicated that certain of the recommendations outlined were worthy of careful consideration and would be referred to the Council for further action.

The next committee report was presented by the chairman of the Meetings Committee, Albert Buenger, St. Paul, Minn.

Report of the Meetings Committee

The functions of the Meetings Committee during the past year have been carried out as provided for in the By-Laws and resulted in the presentation of the papers at the Summer Meeting at Toronto in 1935 and at the 42nd Annual Meeting here in Chicago this week, which papers are a matter of record.

After a slow start the Committee finally got under way and with the invaluable help of Mr. Hutchinson the program for the Toronto Meeting was assembled.

After the Summer Meeting, papers began to come in and the notice sent out by the Secretary seemed to stimulate the production of papers. By the early fall it was evident that sufficient papers would be available for the Annual Meeting.

At the suggestion of President Howatt one of the papers from the Research Laboratory was circulated among the Chapters for presentation and discussion. From the reports we have received and from the experience in our own Chapters, it was evident that this arrangement proved to be very popular as well as profitable. We

would seriously recommend that this arrangement be continued in the future.

One of the problems which presented itself during the year concerned the production of research papers. We should like to quote from a letter received from the Chairman of the Research Committee:

"I agree with you that the idea of having a research paper ready in time for discussion by the Chapters in the Fall is a good one. The only trouble with this is that the men doing the research are continuously being pushed to get out papers ready for the meetings. The result is that they are forced to work right up to the deadline in order to get in under the wire. This pressure I am afraid is sometimes going to lead to the presentation of half-baked results."

At the end of the year some papers have accumulated which can be presented at future meetings. We would therefore recommend that a policy be established to have at all times enough papers on reserve to prevent the crowding of the Research Laboratory staff and the last minute scramble for papers.

In closing we want to express our appreciation for the material assistance of A. V. Hutchinson and the staff at the New York office rendered in carrying out the work of the Committee.

Respectfully submitted,

ALBERT BUENGER, *Chairman*,
F. E. GIESECKE,
O. W. OTT,

MEETINGS COMMITTEE.

President Howatt declared that this Committee report should be included in the Proceedings of the meeting.

Report of the Committee on Research

The Report of the Committee on Research was prepared by Prof. A. P. Kratz, Chairman, and F. C. Houghten, Director of the Research Laboratory, and was presented in abstract by Professor Kratz.

The Committee on Research of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS held four meetings during the year. At the first meeting, held in Buffalo on January 28, a budget for 1935 was prepared and other business pertaining to the 1934 committee was completed. The 1935 committee met immediately following adjournment of the 1934 committee, approved the budget recommended by the previous committee, and adopted a research program. Two meetings of the Committee were held in Toronto, Ontario, during the Semi-Annual Meeting of the Society, at which time the work of the various Technical Advisory Committees and the studies carried on under them at the Laboratory in Pittsburgh and in the cooperating universities were reviewed.

The Committee operated under a budget somewhat larger than the previous year, and found no great difficulty in meeting its requirements. The funds on hand at the close of the year are greater than those on hand when the Committee was organized.

Sixteen Technical Advisory Committees served as sub-committees of the Committee on Research, each to consider a separate technical subject which it studied or outlined for laboratory investigation. Three new Technical Advisory Committees on (1) Summer Cooling Standards, (2) Frictional Resistance to Flow of Air in Small Ducts and Fittings, and (3) Corrosion in Air Conditioning Equipment, were appointed, and the work of determining summer cooling standards was placed on the Research Laboratory's program for immediate investigation.

An extensive program of research was carried on at the Laboratory in Pittsburgh and in cooperation with nine universities. These activities resulted in 11 technical papers presented to the Society, and the development of a code on Garage Ven-

tilation which was formally adopted as a standard of the Society by letter ballot of the membership.

The Committee on Research has always considered it desirable to keep in contact with the Society through the various chapters, and also to keep in contact with other organizations who should be interested in its work. It regretted the necessity of curtailing this activity during the depression because of insufficient funds. With improved conditions during the current year it was possible to re-establish this practice in a small way. The Chairman of the Committee attended three chapter meetings and the Director of the Laboratory attended two chapter meetings at which the work of the Committee was discussed. Contacts with other technical groups were continued wherever possible. It is hoped that this activity may be increased in the future.

EXHIBIT III

RESEARCH LABORATORY AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

STATEMENT OF INCOME AND EXPENSES

Year Ending December 31, 1935*

Research Fund (Unrestricted)

INCOME			
Contribution from General Fund as provided in By-Laws—40% of Dues Collected from Members and Associate Members			$ 9,172.76
Appropriations:			
From General Fund		$1,500.00	
From Endowment Fund		662.50	
			2,162.50
Contributions from Outside Sources			5,798.36
Interest on Bank Deposit			68.00
			$17,201.62
EXPENSES			
SALARIES			
Laboratory Staff	$6,620.00		
Graduate Student Help	1,187.00	$7,807.00	
Cooperative Research Contracts		6,148.36	
Special Fellowship at University of Illinois Medical School		450.00	
Correlating Thermal Research		500.00	
Laboratory Supplies and Equipment		438.50	
Travel		501.88	
Printing and Promotion		215.00	
Exhibits and Meetings		135.36	
Office Supplies, Equipment and Expense		311.08	
			16,507.18
Excess for Year of Income over Expenses of Research Fund (Unrestricted)			$ 694.44

* Accompanying report of Price, Waterhouse & Co., dated January 21, 1936.

1935 Research Contributors

Contributing Funds

Armstrong Cork Products Co.
Crane Co.
C. A. Dunham Co.
Heating, Piping and Air Conditioning Contractors National Association
Modine Manufacturing Co.
Penn Electric Switch Co.
Portland Cement Association
Trane, Co., The
A. C. Willard.

Contributing Equipment

Aerofin Corp.
American Radiator Co.
Detroit Lubricator Co.
Julien P. Friez and Sons, Inc.
Frigidaire Div., General Motors Corp.
General Electric Co.
Illinois Testing Laboratories, Inc.
Johnson Service Co.
Meyer Furnace Co., The
Minneapolis-Honeywell Regulator Co.
National Warm Air Heating and Air Conditioning Association
John J. Nesbitt, Inc.
Weil-McLain Co.
York Ice Machinery Corp.

Universities Cooperating with the Committee on Research

1. AGRICULTURAL & MECHANICAL COLLEGE OF TEXAS: Heat Requirements of Buildings.
2. CASE SCHOOL OF APPLIED SCIENCE: Heat transfer from and to direct and extended surfaces with forced air circulation.
3. CORNELL UNIVERSITY: Heat requirements of buildings.
4. HARVARD SCHOOL OF PUBLIC HEALTH: Minimum air requirements for ventilation.
5. UNIVERSITY OF ILLINOIS: (1) Direct and indirect radiation with gravity air circulation. (2) Cooling of buildings in summer. (3) Air conditioning in the treatment of diseases.
6. MICHIGAN COLLEGE OF MINING AND TECHNOLOGY: Corrosion in return lines in relation to the chemical composition of the water, vapor and gas handled.
7. MARQUETTE UNIVERSITY: Methods of air supply and distribution in air conditioning.
8. UNIVERSITY OF MINNESOTA: (1) Dust and dust control apparatus. (2) Heat transmission through concrete walls.
9. UNIVERSITY OF WISCONSIN: Heating of buildings.

Papers Resulting from Research During 1935

Eleven technical papers, resulting from research carried on in Pittsburgh or in the cooperating universities under the several Technical Advisory Committees were prepared for presentation at meetings of the Society and publication by the A. S. H. V. E. They are listed below and will be referred to by number in the discussion of research projects later in this report.

1. A Carbon Monoxide Alarm and Ventilation Control, by F. C. Houghten and Linwood Thiessen, A. S. H. V. E. TRANSACTIONS, Vol. 41, 1935.
2. Classroom Odors with Reduced Outside Air Supply, by F. C. Houghten, H. H. Trimble, Carl Gutberlet and Merle F. Lichtenfels, A. S. H. V. E. TRANSACTIONS, Vol. 41, 1935.
3. The Dust Problem in Air Conditioning, by F. B. Rowley, A. S. H. V. E. TRANSACTIONS, Vol. 41, 1935.
4. Air Supply to Classrooms in Relation to Vent Flue Openings, by F. C. Houghten, Carl Gutberlet and Merle F. Lichtenfels, A. S. H. V. E. TRANSACTIONS, Vol. 41, 1935.
5. Oil Burning in Residences, by D. W. Nelson, A. S. H. V. E. TRANSACTIONS, Vol. 41, 1935.
6. Classroom Drafts in Relation to Entering Air Stream Temperature, by F. C. Houghten, H. H. Trimble, Carl Gutberlet and Merle F. Lichtenfels, A. S. H. V. E. TRANSACTIONS, Vol. 41, 1935.
7. Comfort Standards for Summer Air Conditioning, by F. C. Houghten and Carl Gutberlet, A. S. H. V. E. TRANSACTIONS, Vol. 42, 1936.
8. Performance of Fin-Tube Units for Air Heating, Cooling and Dehumidifying, by G. L. Tuve, A. S. H. V. E. TRANSACTIONS, Vol. 42, 1936.
9. Thermal Properties of Concrete Construction, by F. B. Rowley, A. B. Algren and Clifford Carlson, A. S. H. V. E. TRANSACTIONS, Vol. 42, 1936.
10. Ventilation Requirements, by C. P. Yaglou, E. C. Riley and D. I. Coggins, A. S. H. V. E. TRANSACTIONS, Vol. 42, 1936.
11. Corrosion Studies in Steam Heating Systems, by R. R. Seeber, F. A. Rohrman and G. E. Smedberg, A. S. H. V. E. TRANSACTIONS, Vol. 42, 1936.

Research Projects Investigated During 1935

During the year 16 projects were given consideration under as many Technical Advisory Committees. Because of the budget limitations, laboratory research, either at the Laboratory in Pittsburgh or at the cooperating universities, was possible in only a few cases; however, the Technical Advisory Committees were in most cases successful in giving consideration to the subject in hand with measured progress. The personnel of each of the Committees, working under the Committee on Research, and the work carried on under them during the year are given.

1. AIR CONDITIONS AND THEIR RELATION TO LIVING COMFORT.—*Technical Advisory Committee:* C. P. Yaglou, *Chairman;* J. J. Aeberly, W. L. Fleisher, R. R. Sayers, C.-E. A. Winslow.

The committee has continued a study of the minimum ventilation requirements for health and comfort. The cooperative laboratory investigation of the subject was carried on at Harvard School of Public Health. This work will be continued another year if resources allow. A progress report [1] is to be presented at the Annual Meeting, giving ventilation requirements of various groups of school children and adults under representative practical conditions. The report shows clearly the impossibility of fixing any single standard that would apply under all conditions. Each case, it would seem, must be considered on its own merits. The important factors appear to be personal sanitation, air space allowed per person, and odor removing capacity of air conditioning processes. The range in outdoor air requirements is from 5 cfm per person, or less, when the air is completely conditioned, to 38 cfm in the case of grade school children of the poorest class, when the air is not conditioned.

Considerable evidence is adduced to the effect that there are other factors in the ventilation of occupied rooms, besides temperature and humidity which affect the quality and pleasantness of air from the point of view of both occupants and visitors. This evidence confirms from a different angle an old conclusion of the New York State Commission on Ventilation, and also the very latest finding of Winslow on *weather feelings.*

Another activity of the committee during the past year was a critical review of the literature on carbon dioxide, humidity and ozone, supplemented by a number of tests at Harvard, for the purpose of finding out where information is inadequate and whether there is sufficient justification for additional research.

The Research Laboratory in Pittsburgh reported a brief study [2] of the odor conditions resulting with reduced ventilation in classrooms in connection with a study of school room ventilation under the Technical Advisory Committee on Minimum Temperature and Method of Introduction of Cooling Air in Classrooms.

2. AIR CONDITIONING IN THE TREATMENT OF DISEASES.—*Technical Advisory Committee:* E. V. Hill, *Chairman;* N. D. Adams, J. J. Aeberly, Margaret Ingels, H. R. Linn, P. J. Marschall, E. L. Stammer.

This committee was appointed a year ago to develop cooperation with medical schools in a study of the relation of air conditioning to the treatment of diseases. The committee has been active in giving consideration to the problem throughout the year and was successful in developing a cooperative arrangement with the College of Medicine of the University of Illinois in Chicago, under which a comprehensive study will be carried on. Complete details of the testing program have not been decided upon at present, but it is proposed first to study a group of normal individuals when subjected to various atmospheres with temperatures ranging from 50 to 100 F and relative humidities ranging from 20 per cent to 60 per cent or more.

A number of different studies have been proposed to measure the body's behavior under these atmospheric conditions. These are:

(1) Mental tests involving various functions and determined somewhat by the subject's educational background.

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[1] See No. 10 in List of Research Papers, p. 13.
[2] See No. 2 in List of Research Papers, p. 13.

(2) Circulatory tests giving the state of blood circulation under basal conditions consisting of complete physical and mental relaxation, a period of 12 to 18 hours post-prandial, and freedom from all drugs. The following tests would be made:
   a. Metabolic rate, including respiratory quotients.
   b. Oxygen saturation of arterial blood and oxygen capacity.
   c. Analysis of alveolar air, vital capacity. Study of character of respiration.
   d. Circulation times.
   e. Cardiac output.
   f. Bloob volume blood counts; total fluid content.
   g. Reaction of skin to salt solutions, an index of its hydrophilic properties.
   h. Capillary and venous pressure.
   i. State of capillaries and smaller blood vessels as tested by the plethysmograph and skin temperature.

(3) Studies of appetite, gastro-intestinal behavior, and hunger contraction might be added.

Either subsequent to or at the same time that studies are made on normal patients it is planned to study the effect of air conditions on patients having the following circulatory disorders:

(1) Toxic goiter.
(2) Rheumatic heart which is on the border line of decompensation.
(3) Rheumatic heart which is compensated.
(4) Compensated hypertensive heart.
(5) Hypertensive heart on the border line of decompensation.

3. ATMOSPHERIC DUST AND AIR CLEANING DEVICES.—*Technical Advisory Committee:* H. C. Murphy, *Chairman;* J. J. Bloomfield, M. I. Dorfan, Philip Drinker, Leonard Greenburg, S. R. Lewis, F. B. Rowley, D. C. Simpson, W. O. Vedder.

Since the last Annual Meeting the Committee has continued its studies and investigations looking to the development of a code for testing and rating air cleaning devices used in dust hazardous occupations. Much work has been conducted by correspondence, with such incidental meetings as were possible at the National Safety Congress and similar engineering assemblages, and through cooperation between the Director of the Laboratory and the U. S. Bureau of Mines, as an aid in the development of a code.

The Chairman spent some time in England during the past Summer, and had an opportunity of studying the work being done there in correcting atmospheric pollution. The officers and members of the *Institution of Heating and Ventilating Engineers of Great Britain* were courteous and cooperative, and gave every possible assistance in these investigations.

Many considerations recommended further careful and detailed study before recommendations for the proposed code are submitted to the Society, and it is hoped that definite procedures can be submitted for consideration at the June Meeting.

4. COMFORT STANDARDS FOR SUMMER COOLING.—*Technical Advisory Committee:* W. L. Fleisher, *Chairman;* A. E. Beals, F. R. Bichowsky, Elliott Harrington, E. V. Hill, C. P. Yaglou.

This is a new committee appointed during the year to undertake a study of comfort standards for summer cooling. A great deal of dissatisfaction was found by the committee in the profession and industry with the present standards as contained in the A. S. H. V. E. GUIDE.

A preliminary survey of the entire subject was outlined by the Committee and carried out by the Research Laboratory during the summer months. While this study was in no respect conclusive as establishing finally acceptable standards, it accomplished a great deal in indicating what may be expected as a result of further study. A paper resulting from this work is being presented [3] at the Annual Meeting.

Additional work was carried on by the Chairman of the committee, dealing with possible rise in temperature with higher air velocities and maximum relative humidities along effective temperature lines outside the so-called comfort zone.

The studies so far made have not only served to conclusively demonstrate that the

[3] See No. 7 in List of Research Papers, p. 13.

present standards are unnecessarily limiting, but they have further served to point out the need for thorough investigation of several phases of the subject. Among the different factors which may possibly affect the final establishment of summer cooling standards and which are recommended by the committee for further study are the following:

(1) A more definite upper limitation in allowable relative humidity at the proper effective temperature.
(2) Variations in the desirable effective temperature based upon the daily prevailing outside condition from which persons enter a conditioned room.
(3) Variations in the desired indoor effective temperature based upon the average climatic condition outdoors, based not on daily change but upon an average over a much longer period of time, or in other words, the variation in desired indoor effective temperature based upon the summer climatic condition for a given city or region; usually fixed by latitude.
(4) Variations in the indoor effective temperature desired by persons of varying sensitivity, including older persons and persons with lower vitality.
(5) A study of the conditions which may be interpreted by a person as a draft, as determined by excessive air motion at the prevailing indoor temperature, and for air streams of other temperatures, usually streams of entering cooler air which may strike a person.
(6) A study of the physiological reactions of a person before, upon entering, during occupancy, immediately upon leaving, and over an hour or two after leaving a cooled room, particularly observing the changes in skin temperature on various parts of the body, and perspiration.

A discussion of these items is given.

(1) *A More Definite Upper Limitation in Allowable Relative Humidity:* A series of tests should be made in which the main purpose would be to determine the relative effect on persons entering a cooled room at the same effective temperature from the same outside conditions, when the relative humidity or moisture content of the cooled room varies. The study of last summer indicated no variation whatever for relative humidities as high as 70 per cent, and no outstanding variation for relative humidities as high as 85 per cent. The only way in which such effects can be determined satisfactorily is to make a series of tests with sufficient number of the variables controlled so that the variations due to this one factor may be determined. Perhaps 12 to 15 or 20 tests would be conclusive in settling this point.

(2) *Variations in the Desirable Effective Temperature Based Upon the Daily Prevailing Outside Conditions:* Last summer's study indicated no marked or outstanding variations desired by the subjects. However, since other variables, such as the indoor relative humidity, length of exposure, degree of perspiration, etc., were not controlled, the data were not voluminous enough to draw any closely determined conclusions regarding possible small variations in the desired indoor effective temperature.

(3) *Effective Temperature Variations Desirable for Indoor Cooling for Regions of Different Climatic Conditions:* The study made last summer could not in any sense answer this question, which is of interest to a great many air conditioning engineers who operate throughout different geographical locations of the country. That the summer climatic condition of a given region must necessarily affect the desired indoor conditions for summer cooling may be accepted as axiomatic, from the fact that for winter heating an effective temperature range of from 63 to 71 deg is desired, while for summer cooling an effective temperature range of from approximately 71 to 74 was indicated for the Pittsburgh locality during last summer Certainly with less severe summer conditions than pertaining in Pittsburgh, summer cooling would require some intermediate effective temperature range. It might be argued with equal logic, that for regions having more severe summer climates, as an example, New Orleans, La., or Dallas, Tex., a higher effective temperature range for indoor summer cooling would be required.

One significant fact brought out in last summer's study, however, leads to the belief that the desirable indoor effective temperature range for summer cooling may not go any higher than that found for Pittsburgh, in spite of higher outside temperatures. The desirable effective temperature range for indoor cooling in Pittsburgh last summer was found to have an upper limit but a degree or two lower than the perspiration line. Since it may be accepted as axiomatic that summer cooling conditions must not result in sensible perspiration, it would follow that the upper limit of the effective temperature range for any region could not go higher, unless

the perspiration line also moves upward with different climatic conditions. Perspiration being a fundamental physiological reaction, it would seem unlikely to occur at varying effective temperatures based upon climatic conditions, although this fact has not been established. This phase of the subject should be next studied by selecting a minimum of two additional localities, one in a hotter region near the Gulf Coast, and one in a cooler region, such as Toronto, Ontario. If two such localities as New Orleans and Toronto should be studied next summer and the results compared with those found at Pittsburgh, it should be possible to more definitely establish the fact that climatic variations require variations in desired effective temperatures for summer cooling. If Pittsburgh and these other two localities show little variation, then the subject could be dropped; on the other hand, if these three localities should show a wide variation, then it might be desirable to extend the study to other localities having different summer climates.

(4) *Variation in Age and Sensitivity of Persons in Relation to Summer Cooling:* Undoubtedly, the conclusiveness of the Laboratory's study last summer can be criticized most because of the fact that the subjects were young, rugged men. In this connection, it is well to point out that this criticism was anticipated before last summer's study was made, and that the criticism of the Laboratory's work last summer is not so much the finding of fault in the plan followed by the Laboratory as it is a criticism of using the results as conclusive until further work is done. It was agreed that it was desirable to make the preliminary survey with persons not supersensitive to summer cooling, and then later to study variations from these basic findings. It is certain, however, that additional work should be done to show variations resulting from variations in age and sensitivity in individuals. Those associated closely with the work are inclined to believe that those conditions which the young subjects classed as *comfortable*, or at (4) in the test scale, will not be uncomfortable to most older and more sensitive people. It is thought that the difference will rather lie in the ability of different age groups to stand lower temperatures without undue discomfort. As an example, it may be expected that all age groups will be comfortable in the same effective temperature range in which the young subjects were comfortable, but that while the young subjects experienced no great discomfort at effective temperatures of 70 deg and lower, but simply voted that they felt *cool*, older persons will not only feel *cool* but they will feel decidedly uncomfortable. This is, of course, only a conjecture, and further study will prove it either true or false. If, however, this conjecture is true, this phase of the problem can be easily disposed of; on the other hand, if it is not true and it is found that different age groups desire different effective temperature ranges, then a final solution of the problem will be difficult to find.

(5) *Drafts in Cooling:* The question of what constitutes objectionable drafts is always a vital question when cooled air is admitted to a room, or when cooling is applied otherwise. Considerable work has been done on what constitutes allowable air motion within occupied space in heating practice. Many are of the opinion that these same limitations apply to summer cooling. It is almost certain that they do not apply to the same extent. In fact, it is probable the air movement in summer cooling can be used as a considerable factor in reducing the sense of warmth. To be so used, however, air movement must necessarily be uniform so as not to produce excessive cooling of any localized parts of the body. While, as mentioned above, there is considerable data on the subject of allowable air velocities in occupied space at, for example, 70 F during the heating season, these velocities must be uniform and of the same temperature as the uniform room temperature.

There is perhaps as little data available on the subject of what constitutes a draft in heating practice as in cooling practice. Therefore, aside from the question of determining what are allowable uniform air movements in a room for cooling when this air velocity is accounted for in the effective temperature, it is desirable to make a study of cold drafts as such, both for such temperatures as are used in heating and in cooling. This problem is one and the same, and should be carried on as applying to both heating and cooling. The greatest difficulty that the Laboratory had in its study of Minimum Temperature and Method of Introduction of Cooling Air in classrooms last winter was in determining what a draft was. Undoubtedly, a draft is one of the most discussed phenomena considered by the heating, ventilating

and air conditioning engineer, and yet, to date there has been no effort to determine what combinations of air movement and temperature constitute a draft.

Since this phase of the study is most fundamental to the entire heating, ventilating and air conditioning industry, as well as to the study of summer cooling standards, it should be carried on at once. At least most phases of the study can be carried on without respect to winter or summer, and it is suggested that the Laboratory proceed with this phase of the study at once. In making the study, means should be set up in the psychrometric rooms for producing a localized area of rather constant, linear air movement, the temperature of which can be controlled, irrespective of the surrounding more or less uniform room temperature. Arrangements should be such that this uniform air velocity of known temperature and speed may be directed against various parts of the body, including the top of the head, the back of the neck, the back, the thigh and the ankles. The study probably will not be difficult nor require a great deal of time.

(6) *Physiological Effects on Entering, Occupying and Leaving a Cooled Room:* In order to have a better understanding of what takes place when a person receives a shock upon entering a cooled space or a *relapse* upon reentering the hot outside, it is desirable to make a brief study of such physiological reactions as skin temperature and perspiration effects during such changes. Many persons are talking about skin temperature effects without knowing much about what they are and the variations met with under the changes referred to previously. This phase of the study will not be long, and at least a large part of it can be carried on either during the winter or summer. In fact, some of the results may be gleaned from a further analysis of data available at the Laboratory and collected several years ago.

(7) *Physiological Reactions to Air Conditions Which May be of Value in the Treatment of Diseases:* During the past few months the Laboratory, working under this committee, has developed a friendly cooperation with medical staff at the University of Pittsburgh. The committee is particularly happy over the fact that these members of the medical profession came to the Laboratory with their problem, as a result of the Laboratory's papers of a physiological nature published over the past several years.

These members of the medical staff are giving particular attention to the treatment of certain diseases by subjecting the individual to certain definitely controlled atmospheric conditions, particularly pertaining to atmospheric temperature. A few years ago the Laboratory made its psychrometric rooms available for the treatment of a few syphilitic patients. As a result of this study, it was rather conclusively proven that syphilis could be cured by subjecting the patient to a high temperature atmosphere. This cure seemed to be equivalent to the prevailing cure by giving the patient malaria fever. Probably the only advantage in the malaria fever treatment was the high body temperature resulting. Following these disclosures, the question arose as to whether the high body temperature killed micro-organisms within the body directly, or whether the high temperature brought the effect about indirectly by increasing the white blood cell or leucocyte count of the blood, which is recognized as a measure of the ability of the body to throw off infections.

The Laboratory recently cooperated with the medical staff at the University of Pittsburgh in making the psychrometric rooms available for studies of the effect of high temperature of the body on its blood count, and in which members of the Laboratory personnel acted as subjects. It was found that upon raising the body temperature 104 F in a saturated atmosphere of 110 F, the leucocyte count of the blood was, during the same period of time, increasing from about 5,000 to about 14,000, hence definitely establishing the fact that the high temperature does immediately change the leucocyte count of the blood.

Whether or not these developments will have far reaching effects in the treatment of diseases by air conditioning remains to be seen. However, it at least opens up a very extensive and most interesting range of possibilities. The Laboratory has been asked to collaborate with the University in preparing a paper on the brief findings mentioned.

5. CORROSION IN AIR CONDITIONING EQUIPMENT.—*Technical Advisory Committee:* A. E. Stacey, *Chairman;* M. S. Kice, F. N. Speller, C. M. Sterne, R. T. Thornton, J. H. Walker.

This committee was appointed in the late summer to undertake a study of corrosion problems in air conditioning equipment. The committee has since been actively engaged in formulating a program. While a complete program has not yet been completed, the committee is at the present time studying the seriousness of the corrosion problem in general, the prevailing conditions surrounding locations of serious corrosion, the metals involved, and whether the corrosion results from contact with the atmosphere or water. The committee is giving further consideration to atmosphere contamination which may result in corrosion, the characters of water which may result in corrosion of metals in contact with it, and possible treatment of such water in order to improve conditions. Consideration is also being given to the effectiveness of protective coatings in reducing corrosion. In studying the seriousness of the corrosion problem in order to determine whether research is desirable, the committee is giving thought to air conditioning equipment which may be included in several different types, which based upon usage may react differently as regards corrosion.

6. CORROSION IN HEATING SYSTEMS.—*Technical Advisory Committee:* J. H. Walker, *Chairman;* E. L. Chappell, W. H. Driscoll, C. A. Dunham, L. B. Miller, R. R. Seeber, C. M. Sterne.

This committee was organized to cooperate with the Michigan College of Mining and Technology in a study of corrosion of ferrous metals in heating systems. A research project has been under way at that institution since June, 1933. A preliminary report was presented to the 1934 Annual Meeting by Prof. R. R. Seeber. A second report[4] will be presented at the Semi-Annual Meeting. The results give promise of throwing considerable light on the knowledge of corrosion in heating system return lines and methods of reducing it. Up to the present time the work has been financed entirely by the college, but it is hoped to obtain financial support from the industry during 1936.

While the committee was originally organized to study corrosion of ferrous metals only, in heating systems, there has been a persistent demand that it also give consideration to corrosion of copper and copper bearing alloys in such systems. Consideration was given to this phase of the subject during the year. A questionnaire was sent out to a number of persons interested in the subject, and later this information was surveyed and discussed at a conference during the Semi-Annual Meeting. At this conference there was considerable interest shown in the problem, but there was no unanimous agreement concerning the need of such a study, it being contended by some that the corrosion resulting from the use of copper and copper bearing alloys in heating systems was so slight in comparison with corrosion of ferrous metals as to be negligible. Others reported considerable trouble from such corrosion, while others argued that the decreased thickness of copper and copper bearing alloys used might more than offset the decreased rate of corrosion. Another conference on the subject was arranged for and held in New York on Tuesday, December 3, attended by 13 engineers interested in the subject and representing 12 different organizations. No plans for future action were arrived at, but it was agreed that continued consideration of the subject was desirable.

7. DIRECT AND INDIRECT RADIATION WITH GRAVITY AIR CIRCULATION.—*Technical Advisory Committee:* H. F. Hutzel, *Chairman;* M. K. Fahnestock, H. R. Linn, J. P. Magos, J. W. McElgin, J. F. McIntire, T. A. Novotney, R. N. Trane.

The committee continued to give consideration to the subject during the past year and active research was continued at the University of Illinois, in spite of limited funds. Two investigations are being made, neither of which is completed, but final reports may be expected for the Semi-Annual Meeting.

[4] See No. 11 in List of Research Papers, p. 13.

The work in the warm wall test booth consists of a study of the inlet and outlets of convector heater cabinets with reference to their effect upon the heat output or capacity of the units, in an attempt to arrive at definite recommendations as to the optimum dimensions for such openings in terms of the free area through the heating unit or the cross-section area of the cabinet. Three convectors, each having a different type of heating unit, are being tested. This program is approximately 30 per cent completed and the results indicate that there is an optimum inlet and outlet arrangement for a given cabinet and heating unit combination and that a variation from this optimum, especially in the case of the outlet, will materially reduce the heat output.

In the room heating testing plant an investigation is being made to determine the rate at which a room cools and heats with different types of radiators and convectors. This involves the effect on room air temperatures of the residual heat stored in cast-iron and non-ferrous convectors and cast-iron radiators when the steam supply is shut off. Four convectors, two having cast-iron heating units and two having non-ferrous heating units, and two types of cast-iron radiators are being tested. This investigation is nearly completed and it is anticipated that the results will be presented at the Semi-Annual Meeting.

Immediately following the completion of the program it is planned to undertake a brief study of the effect of insulating, first, the ceiling and then both the exposed walls and ceiling, of the test room in the room heating testing plant. This study will involve a comparison of the steam condensation and the room temperature conditions with the room uninsulated, with the ceiling insulated, with the ceiling uninsulated and with both walls and ceiling insulated.

8. EFFECT OF ENTERING TEMPERATURE AND VELOCITY ON THE TEMPERATURE AND DISTRIBUTION OF AIR WITHIN AN ENCLOSURE.—*Technical Advisory Committee:* C. H. Randolph, *Chairman;* E. H. Baars, J. S. Jung, F. A. Kartak, J. E. Schoen, Ernest Szekely, J. H. Volk.

The committee completed plans for the study to be carried on in cooperation with Marquette University at Milwaukee. Suitable space was made available and a few preliminary tests were made.

9. FRICTIONAL RESISTANCE TO FLOW OF AIR IN SMALL DUCTS AND FITTINGS.—*Technical Advisory Committee:* J. H. Van Alsburg, *Chairman;* C. A. Booth, S. H. Downs, L. B. Miller, L. E. Smith.

This committee was appointed during the present year to make a study of the frictional resistance to the flow of air in ducts, elbows and other fittings of small size, such as are finding wide application at the present time in air conditioning and heating systems in small buildings, and particularly as used in fan-furnace systems.

The committee has given considerable study to the problem and has outlined a program to be followed by the Laboratory in Pittsburgh, including the building up of a test set-up in which the frictional resistance to the flow of air may be studied in ducts and fittings of various sizes and design. This work is now being actively taken up by the Laboratory.

10. GAS HEATING—*Technical Advisory Committee:* W. E. Stark, *Chairman;* R. M. Conner, C. A. Dunham, Robert Harper, E. A. Jones, Thomson King, J. F. McIntire, E. L. Tornquist, H. L. Whitelaw.

This committee is continuing to give consideration to problems connected with the use of gas in central heating systems. No projects have been outlined requiring laboratory study beyond those being investigated by the testing laboratory of the *American Gas Association.*

11. HEAT REQUIREMENTS OF BUILDINGS.—*Technical Advisory Committee:* D. S. Boyden, *Chairman;* P. D. Close, W. H. Driscoll, H. M. Hart, P. E. Holcombe, V. W. Hunter, G. L. Larson, H. H. Mather, E. C. Rack, F. B. Rowley, R. J. J. Tennant, J. H. Walker.

The Committee on Heat Requirements of Buildings, appointed two years ago for the purpose of developing a more rational understanding of the factors affecting the heating requirements of buildings, has continued to be active during the past year. Although several research projects dealing with certain phases of the subject are being studied, their effort in arriving at a final solution of the problem has proven extremely slow, largely for the reason that a study in a given building under definitely prescribed conditions must be continued throughout an entire heating season to be of any value whatever. Further, no one such study can be relied upon as a basis for a rational understanding of the complicated problem, but must rather be considered as one link in a long chain of evidence which must be completed before a final solution and conclusive report can be made by the committee. Invariably, however, all the evidence being analyzed by the committee tends to indicate that our present methods of estimating heat losses from buildings give values greater than those established by test. The real purpose of this committee is to eliminate the discrepancy. In attacking its problem, the committee has had the close cooperation and support of the *National District Heating Association*, the *American Gas Association*, the *National Warm Air Heating and Air Conditioning Association*, and others interested in the heating requirements of buildings.

During the life of the committee the following reports, resulting from definite concrete studies, have been presented, each of which has been helpful in solving the committee's problem:

1. Selecting Temperatures and Wind Velocities for Calculating Heat Losses, by P. D. Close. A. S. H. V. E. TRANSACTIONS, Vol. 40, 1934.
2. Influence of Stack Effect on the Heat Loss in Tall Buildings, by Axel Marin. A. S. H. V. E. TRANSACTIONS, Vol. 40, 1934.
3. Wind Velocities Near a Building and Their Effect on Heat Loss, by F. C. Houghten, J. L. Blackshaw and Carl Gutberlet. A. S. H. V. E. TRANSACTIONS, Vol. 40, 1934.
4. Tests of Three Heating Systems in an Industrial Type of Building, by G. L. Larson, D. W. Nelson and John James. A. S. H. V. E. TRANSACTIONS, Vol. 41, 1935.
5. Heat Requirements of Buildings, by J. H. Walker and G. H. Tuttle. A. S. H. V. E. TRANSACTIONS, Vol. 41, 1935.

During the current year the committee has had several projects under consideration. In some cases these projects have resulted in reports for presentation at the Society's meetings, while in other cases, the projects have reached various stages of completion. These will be discussed as follows:

Heating Requirements—Sibley Dome, Cornell University: The committee, working with the Laboratory in Pittsburgh, cooperated with the Cornell University through H. M. Ward of the Department of Buildings and Grounds, and Perry West, acting as consultant for the University, and perfected plans for a comprehensive study of the various factors affecting heat losses from Sibley Dome, one of the buildings of the University. Data were collected under Mr. Ward's direction throughout the past heating season, giving a daily record of the steam consumption together with temperatures in various parts of the building, the character of building occupancy, and the outside temperature and wind velocity observed in close proximity to the building. These data are being analyzed and a paper is being prepared for presentation to the Society. This study and the resulting report should add much to knowledge of the variation in the daily heating load with occupancy and outside weather conditions for a typical university building.

Heat Requirements—Agricultural and Mechanical College of Texas: Plans have been perfected through the committee and the Laboratory, in cooperation with Prof. F. E. Giesecke, for a study of the heat requirements of one of the buildings at the Agricultural and Mechanical College of Texas. The active collection of the data in this study was started during the fall and will be continued throughout the present heating season.

The building under test is heated with forced circulation hot water. Recording

and integrating water flow meters located in the supply to, and return from, the building, together with recording thermometers, will give an accurate record of the heat delivered to the building. Recording thermometers located throughout the building will give the air temperatures maintained, which taken together with the construction characteristics of the building and the outside temperature, wind velocity and solar radiation will allow a close correlation between our available knowledge of heat transfer and heat capacity of the walls and the actual heat loss under normal building operation and weather conditions.

Heat Requirements—Grant Building, Pittsburgh, Pa.: The Research Laboratory has made a study of the hourly heat load required to maintain uniform temperatures in eight offices, having three exposures and on three different floors of the Grant Building, during the last half of the 1933-34 heating season. This study has already resulted in the paper, Wind Velocities Near a Building and Their Effect on Heat Loss. Additional data collected in this study are being analyzed and will be presented to the Society at a later date. This study should result in a correlation between the actual heat loss from the several rooms studied and the simultaneously existing temperature and wind velocity, both as reported by the Weather Bureau for the city and as actually observed a short distance outside of each individual room window.

Comparative Heat Loss from Insulated and Non-Insulated Buildings: The committee was recently supplied a mass of tabulated data collected for it, giving the heat requirements in insulated and non-insulated, electrically heated buildings in Mason City, at the Grand Coulee Dam Site, State of Washington. These data will be analyzed with a view of presenting a paper on the subject to the Society.

Degree Day: A further analysis of the degree-day method of estimating heating requirements and its application to various types of buildings and methods of heating is being made.

Study of Heat Flow Through Roofs and Walls of Buildings as Affected by Heat Capacity and Solar Radiation: The Research Laboratory has from time to time studied heat flow through walls with the Nicholls Heat Flow Meter under various natural weather conditions. It has also supplied data on cyclic flow of heat through three typical roof decks resulting from the diurnal change in weather, including solar radiation. It is felt that an additional study of this subject should be made with a view of making the findings more easily applied in the estimation of heating and cooling requirements. This continued study is being considered by the committee.

Study of Conductivity of Concrete and Heat Transfer Coefficients for Various Wall Constructions Containing Concrete: Through cooperation between the Research Laboratory, the University of Minnesota and the *Portland Cement Association,* a study is being carried on by Prof. F. B. Rowley at the University of Minnesota, on the various factors affecting conductivity and heat transfer coefficients for walls containing concrete. The work has been in progress for a year, during which time 30 different types of walls were constructed and tested to determine their thermal properties. These walls contain different types of aggregate such as sand, gravel, crushed limestone, cinders and haydite. Insulating materials have been placed in the cells of the hollow blocks of some of the walls, and in other cases insulating materials have been used as a sheet of uniform thickness over the full area of the wall. A paper [5] covering the results thus far obtained will be presented to the Annual Meeting. A further program has been formulated, including the construction of an additional 15 walls, which will show the effect of different applications of insulating materials. A report covering this part of the work will be published next spring.

12. HEAT TRANSFER OF FINNED TUBES WITH FORCED AIR CIRCULATION.—*Technical Advisory Committee:* F. B. Rowley, *Chairman;* S. H. Downs, H. F. Hutzel, E. J. Lindseth, R. H. Norris, W. E. Stark, G. L. Tuve.

The committee has continued to give consideration to a number of phases of this complicated problem, including the effect of:

[5] See No. 9 in List of Research Papers, p. 13.

(1) Reversing the direction of heat flow; that is, cooling air with same number of degrees temperature difference between the water flowing through the coil and the air, and with the same velocity of water through the coil and the same velocity of air through the heater.
(2) Investigation of the effect of the velocity of the water flowing in the tube on the surface coefficient of heat transfer from water to tube, or vice-versa.
(3) A study of heat transfer in the dehumidification of air by direct expansion of dichlorodifluoromethane in the coils.

A laboratory study is being made by Prof. G. L. Tuve at Case School of Applied Science. Following the publication of the first report on the subject in 1934, 250 one-hour tests have been made and the results have been correlated with data published by other investigators and with the performance tables published by manufacturers of finned tubing. The study so far indicates that while there is available fair knowledge of the laws governing heat transfer to or from finned tubing, the data available to engineers are in some respects inconsistent. More research is necessary, especially in the field of dehumidification, in order to establish a satisfactory basis for uniform methods of rating these units. It is suggested that the Society establish a simple and uniform method for testing any given design of extended surface unit, and to specify certain features in the developing of complete performance tables from such tests. A report [6] giving the results of the study to date will be presented to the Annual Meeting. This report includes a suggested method of establishing ratings.

13. MINIMUM TEMPERATURE AND METHOD OF INTRODUCTION OF COOLING AIR IN CLASSROOMS.—*Technical Advisory Committee:* Perry West, *Chairman;* J. D. Cassell, S. R. Lewis, J. R. McColl, A. J. Nesbitt, G. E. Otis, C.-E. A. Winslow.

The work of this committee as originally outlined was carried out by the Laboratory in schools in the Pittsburgh district. The study resulted in three papers [7, 8, 9] presented at the Semi-Annual Meeting. While these papers were not entirely conclusive as regards all the phases of the subjects studied, they have served to clarify our understanding of the subject, and no further investigations have been planned by the committee.

14. REFRIGERATION IN RELATION TO AIR TREATMENT.—*Technical Advisory Committee:* M. K. Fahnestock, *Chairman;* E. A. Brandt, John Everetts, Jr., E. D. Harrington, E. D. Milener, K. W. Miller, E. B. Newill, F. G. Sedgwick, J. H. Walker.

The studies carried on under the committee in summer cooling in the Research Residence at the University of Illinois were continued through the summer of 1935, using city supply water available at a temperature of 58 F as the cooling medium. With the exception of the cooling coils and the air circulating fan the duct system was the same as in the previous studies made during the summers of 1932, 1933 and 1934. Due to the unusually cool summer it was possible to supplement the artificial cooling required during the daytime with cooling with outdoor air at night for practically the entire season. The amount of cooling was controlled by an *on* and *off* motor valve located in the water supply line to the cooling coils. A common type of thermostat operated the motor valve to maintain a constant temperature of approximately 80 F in the house. For a few tests, when an attempt was made to maintain a constant effective temperature in the house, the thermostat was supplanted with manual operation. Over the entire periods requiring cooling, the air recirculating fan was operated continuously and outdoor air equivalent to approximately one air change per hour was brought into the house for the purpose of ventilation.

The results of the summer's work using the city supply water with a temperature of 58 F indicated:

(1) That the amount of cooling coil surface required added considerable air resistance to the recirculating system, necessitating the installation of a materially larger recirculating fan than that required for heating.

[6] See No. 8 in List of Research Papers, p. 13.
[7] See No. 2 in List of Research Papers, p. 13.
[8] See No. 4 in List of Research Papers, p. 13.
[9] See No. 6 in List of Research Papers, p. 13.

(2) That a dry-bulb temperature of 80 F could be satisfactorily maintained with the water cooling coils.
(3) That although an appreciable amount of dehumidification was obtained it was not sufficient to reduce the relative humidity in the house below 63 per cent and the resulting air conditions could not be regarded as entirely satisfactory.

The work will be continued during the summer of 1936 using water at two different temperatures below 58 F. The mechanical refrigerating equipment, including a condensing unit and water cooler, for cooling water to any desired temperature, is installed in the Research Residence and ready to operate.

The *National Warm Air Heating and Air Conditioning Association* and a number of manufacturers of air conditioning apparatus cooperated in these studies. A paper will be presented at the next Semi-Annual Meeting.

15. SOUND IN RELATION TO HEATING AND VENTILATION.

—*Technical Advisory Committee:* V. O. Knudsen, *Chairman;* Carl Ashley, C. A. Booth, F. C. McIntosh, R. F. Norris, J. S. Parkinson, C. H. Randolph, J. P. Reis, G. T. Stanton.

The committee continued its study of various phases of the subject throughout the year. A meeting was held in New York on December 9, attended by five members of the committee and others. Consideration was given to the problem of setting up a suitable procedure for measuring the noise of equipment in the factory or labora-

ROOM LOCATIONS

Location	Loudness Level in Decibles to be Anticipated[a]		
	Min.	Representative	Max.
Sound Film Studios	10	14	20
Radio Broadcasting Studios	10	14	20
Planetarium	15	20	25
Residence Apartments, etc	25	35	40
Theaters—Legitimate	25	30	35
Theaters—Motion Picture	30	35	40
Auditoriums, Concert Halls, etc	25	30	40
Churches	30	35	40
Executive Offices—Treated Private Offices	25	33	40
Private Offices—Untreated	40	45	50
General Offices	50	55	60
Hospitals	20	50	70
Class Rooms	30	40	50
Libraries, Museums, Art Galleries	30	40	45
Public Buildings—Court Houses, Post Offices, etc	45	55	60
Small Stores	40	50	60
Upper Floors Department Stores	40	50	55
Stores—General—including Main Floor of Department Stores	50	60	70
Hotel Dining Rooms	40	50	60
Restaurants and Cafeterias	55	65	75
Banking Rooms	50	55	60
Factories	60	70	80
Office Machine Rooms	60	70	80

[a] It is assumed that if equipment noise alone is not greater than these levels of background noise at any frequency band, results will be commercially satisfactory. When both are heard together, an increase in total level of 3 db can be anticipated.

VEHICULAR NOISE

Location	Loudness Level in Decibles to be Anticipated[a]		
	Min.	Representative	Max.
Railroad Coach	60[b]	70	80
Pullman Car	55[b]	65	75
Automobile	50	65	80
Vehicular Tunnel	75	85	95
Airplane	80	90	100

[a] See p. 24.
[b] For train standing in station a level of about 45 db is the maximum which can ordinarily be tolerated.

tory. It was felt that such measurements should be so made that the results would enable the manufacturer to predict what level his equipment would set up in actual installations. Among the factors discussed were the type of room to be used, the type of noise meter, the direction of the microphone and its distance from the unit, etc.

Consideration was given by the committee to a table of noise levels representing the weighted average of figures obtained from all available sources on this subject. This table is given.

Consideration was also given to an acceptable improvement in the standard available data on sound in relation to heating and ventilation as contained in THE GUIDE.

The committee agreed that there was need for more research on this subject than could be expected from commercial firms now interested. In particular, information is needed on the relation between factory or laboratory measurements on the total noise given off by a unit and the actual levels set up in installation. The simple theory on this point requires considerably more experimental verification than is now available. Information is also to be desired on the effect of sound absorbent linings in different sizes and length of duct, and in particular on the effect of such linings in elbows and turns. This problem has been the subject of considerable research by several firms, but the information is by no means complete. It is the recommendation of the committee that the Committee on Research consider the desirability of conducting such studies at the Research Laboratory of the Society.

16. VENTILATION OF GARAGES AND BUS TERMINALS.— *Technical Advisory Committee:* E. K. Campbell, *Chairman;* S. H. Downs, T. M. Dugan, E. C. Evans, F. H. Hecht, H. L. Moore, A. H. Sluss.

The committee continued to give consideration during the year to various problems in connection with garage ventilation. The Garage Ventilation Code, prepared by the committee, was presented to the Society for final consideration and was adopted. Consideration was given to a proposal made by persons interested in ozone for its use in garages in order to minimize the harmful effects of carbon monoxide on occupants. From all information available on the probable chemical reactions between carbon monoxide and ozone in the atmosphere, and the effect of carbon monoxide on persons, either with or without the presence of ozone, it is indicated rather conclusively that no benefit could be had from such application. However, in order not to be too arbitrary in the matter, the committee proposed a brief scientific study of the subject at the Research Laboratory in cooperation with the U. S. Bureau of Mines and the U. S. Public Health Service. This proposal has not been accepted by those promoting the idea.

The report of the Guide Publication Committee was then presented by the chairman, Prof. G. L. Larson, Madison, Wis.

Report of Guide Publication Committee

The Guide Publication Committee has incorporated in the 14th edition of THE GUIDE many new and important additions to the Technical Data Section which extends its usefulness to all who are interested in the professional art of heating, ventilating, and air conditioning.

The original conception of the founders of THE GUIDE has been carefully protected in this new volume so as to maintain its continued role as an authoritative and unbiased presentation of long established as well as newly recognized scientific standards of the profession and allied industries.

Because of increasing demands from various users of THE GUIDE, and because of increased application of air conditioning and cooling to all types of industrial processing and comfort installation, four new chapters have been added to the Technical Data Section on the subjects of Refrigeration, Drying, Electric Motive Power, and Railway Air Conditioning.

In addition to this new material, all of the previous chapters have been carefully checked and reviewed, and in some cases the text has been completely rewritten. Information on unit heaters, ventilators, coolers, and air conditioners has been correlated in one chapter and completely revised. New tables of saturated water vapor, monthly degree day values, and a list of maximum city water main temperatures for over 250 representative cities in the United States and Canada are included.

Other chapters which have been amplified with much new material are: Cooling Methods, Fans, Industrial Exhaust Systems, Automatic Fuel Burning Equipment, Radiators and Gravity Convectors, Piping, Fittings, Welding, Water Supply Piping, Water Heating, Electrical Heating, and Air Conditioning for Industrial Processes.

Basic and fundamental data which have been added to THE GUIDE each year since its inception in 1922 have been retained with additions from research sources made necessary by the progress of the heating and ventilating art.

The Problems in Practice which were a major innovation of the last edition of THE GUIDE have been retained with new and practical examples to expand and amplify the text matter of each chapter. Over sixty new illustrations, curves and charts have been added which enhance the value of the descriptive text material.

In connection with the Manufacturer's Catalog Data Section, the Committee has made a more determined effort to carry out the original conception of having the descriptive material as free as possible from comparative statements and superlative terms, and to present therein such condensed technical information concerning modern equipment as will be of practical value to the user of THE GUIDE.

In general, the manufacturers have cooperated fully to make their data useful, informative and serviceable, so that the user can apply it effectively in the selection of materials or equipment to be used in general design.

The Committee appreciates the aid of those far-seeing manufacturers who have assisted in this cooperative enterprise to advance the art of heating, ventilating, and air conditioning which so vitally affects the comfort, happiness and health of everyone. In 1936 it will be possible to distribute THE GUIDE to more than 12,000 users; and it is extremely gratifying that a greater number of manufacturers recognize the effective service rendered by this reference volume, and have increased their advertising space in order to carry the story of their products in greater detail to the thousands who use this book.

In summary, THE GUIDE 1936, compares with the previous edition as follows: the Technical Data Section has been increased 86 pages; Catalog Data, 40 pages; Roll of Membership, 6 pages; and Frontispiece and Preface, 2 pages; making a total increase of 134 pages. The net sales of THE GUIDE 1935 were $21,941.72 and the income from 1936 GUIDE advertising contracts was $25,367.93.

In behalf of the members of the Society and the profession at large, the Guide Publication Committee gratefully acknowledges the cooperative efforts of the following individuals who have given so generously of their time to the development of this 1936 edition:

T. N. Adlam
J. J. Aeberly
O. W. Armspach
C. L. Arnold
C. M. Ashley
W. R. Beattie
E. H. Beling
J. L. Blackshaw
M. G. Bluth
C. A. Booth
Albert Buenger
W. H. Carlton
Sabin Crocker
D. N. Crosthwait, Jr.
J. M. DallaValle
John Everetts, Jr.
J. W. Hertzler
E. L. Hogan
J. H. Holton
F. C. Houghten
Prof. C. M. Humphreys
H. F. Hutzel
C. F. Kayan
R. T. Kern
R. E. Keyes
Prof. V. O. Knudsen
J. W. Kreuttner
C. H. Lankford
L. L. Lewis
Prof. Axel Marin
T. A. Marsh
H. C. Murphy
Prof. D. W. Nelson
P. Nicholls
R. F. Norris
A. J. Offner
O. W. Ott
J. S. Parkinson
G. C. Polk
E. C. Rack
H. G. Rappolt
Prof. T. F. Rockwell
C. Z. Rosecrans
J. O. Ross
S. I. Rottmayer
Prof. F. B. Rowley
S. S. Sanford
Prof. W. M. Sawdon
Prof. L. E. Seeley
C. G. Segeler
H. C. Sharp
J. D. Smith
A. E. Stacey, Jr.
B. Steele
D. J. Stewart
C. A. Thinn
W. D. Turnbull
G. H. Tuttle
J. H. Van Alsburg
H. A. Wagner
A. R. Walker
W. K. Walker
Prof. C. P. Yaglou

The cooperation of A. V. Hutchinson, Secretary, and John James at the headquarters office who assisted in the preparation of THE GUIDE was of invaluable help to the Committee.

The Committee has released this 14th edition of over 12,000 copies with the sincere hope that it will receive the same generous reception that was accorded its predecessors, and that users will find it the most valuable reference volume in their library.

G. L. LARSON, *Chairman,*
GUIDE PUBLICATION COMMITTEE.

President Howatt introduced W. E. Stark, Cleveland, O., Council member and chairman of the committee who represented the Society in the preparation of the proposed Standard Code for Rating and Testing Air Conditioning Equipment and the Standard Code for Rating and Testing Mechanical Condensing Units, sponsored by *A. S. R. E.*

Mr. Stark presented a brief abstract of the codes, which was followed by a lengthy discussion. It was moved that the Society receive the report and approve the work done to date. The motion was regularly seconded and when presented to the members for a vote was unanimously carried.

W. T. Jones, Boston, Mass., presented the Report of the Tellers as follows:

Reports of Board of Tellers

The votes cast for Officers have been tabulated by your Board of Tellers with the following results:

President—G. L. LARSON 488
First Vice-President—D. S. BOYDEN 488
Second Vice-President—E. H. GURNEY 487
Treasurer—A. J. OFFNER 488
Members of the Council—Three-Year Term:
R. C. BOLSINGER 488
S. H. DOWNS 485
W. L. FLEISHER 488
C. M. HUMPHREYS 488

Scattering votes were recorded for Messrs. H. H. Angus and J. M. Frank.

The votes cast for Members of the Committee on Research have been tabulated by the Board of Tellers with the following results:

Members of the Committee on Research—Three-Year Term:

W. A. DANIELSON	488
C. E. LEWIS	488
D. W. NELSON	488
C. TASKER	488
C.-E. A. WINSLOW	488
Total number of legal votes cast	488

Respectfully submitted,

R. H. CARPENTER, *Chairman,*
R. W. RODMAN,
W. WALTER TIMMIS.

President Howatt then stated that in accordance with this report the candidates for various positions in this Society were declared elected to their respective positions and installation of officers would take place on January 30.

W. T. Jones read the result of the ballot reported by the tellers:

Art. C-II, Sec. 7	445 yes; 15 no
Art. C-III, Sec. 4	439 yes; 21 no
Total number of votes cast	500
Total legal votes	460

Respectfully submitted,

R. H. CARPENTER, *Chairman,*
R. W. RODMAN,
W. WALTER TIMMIS.

According to Report of the Tellers President Howatt ruled that the Amendments to the Constitution and By-Laws were passed and would be effective with the regular constitutional procedure of adoption.

The second session of the Society's Annual Meeting was called to order Tuesday, January 28, at 9:30 A.M., by President Howatt who introduced J. H. Milliken, Chicago, president of the Illinois Chapter and general chairman of arrangements for the meeting.

Mr. Milliken extended greetings and spoke of arrangements for the various entertainment functions during the week.

The meeting was honored by the presence of Edward J. Kelly, Mayor of the City of Chicago, who appeared before the meeting to extend welcome and greetings to the members of the Society.

President Howatt opened the third session on Wednesday, January 29, at 9:30 A.M., and announced that one of the progressive steps taken by the Society during the past year was to make arrangements for a Chapter Officers' meeting at the Annual Meeting. F. C. McIntosh, Chairman of this group, presented the various Chapter representatives in attendance.

The fourth session on Wednesday, January 29, at 2:00 P.M., was a joint meeting with the *National Warm Air Heating and Air Conditioning Association* and was held at the Stevens Hotel with Pres. H. T. Richardson of the *N. W. A. H. & A. C. A.* and Pres. John Howatt of the A. S. H. V. E. presiding jointly as chairmen.

Chairman Richardson introduced the President-elect of the A. S. H. V. E., Prof. G. L. Larson, and the secretaries of the two organizations, A. V. Hutchinson of the A. S. H. V. E. and Allen W. Williams of the *N. W. A. H. & A. C. Association.*

Chairman Richardson then presented the gavel to Pres. John Howatt and the first technical paper was given, which was followed by a brief outline of the A. S. H. V. E. research program by Prof. A. P. Kratz.

President Howatt passed the gavel of authority to President Richardson, who then introduced Dr. A. C. Willard to the joint session. F. G. Sedgwick was presented and as Chairman of the Research Advisory Committee of the *N. W. A. H. & A. C. A.* gave a formal report on activities of that committee.

Chairman Richardson then called for the second technical paper on the program.

After calling the fifth session to order on Thursday, January 30, at 9:30 A.M., President Howatt appointed S. R. Lewis as Chairman *pro tem* of the meeting who instructed W. T. Jones, past president of the Society, to conduct the installation of Officers. A. V. Hutchinson, secretary, was appointed marshal and instructed to escort to the rostrum the outgoing members of the Council, R. H. Carpenter, J. D. Cassell, C. V. Haynes, F. C. McIntosh, and L. W. Moon. Then the incoming members of the Council were presented and escorted to the front of the room by Mr. Hutchinson: R. C. Bolsinger, S. H. Downs, W. L. Fleisher and C. M. Humphreys.

The formal installation of officers followed with the President-Elect, Prof. G. L. Larson, Madison, being presented by Prof. F. B. Rowley; First Vice-President, D. S. Boyden, Boston, was introduced by W. H. Driscoll; Second Vice-President, E. H. Gurney, Toronto, was introduced by H. M. Hart, and the Treasurer, A. J. Offner, New York, was introduced by C. V. Haynes.

President Larson presented M. K. Fahnestock, Urbana, who read a proposed revision to the Code for Testing and Rating Concealed Gravity Type Radiation which referred specifically to a change in an exponent from 1.3 to 1.5 for the steam and hot water codes. Mr. Fahnestock moved that the Society proceed with the necessary constitutional method for adoption of these changes and the motion was carried unanimously.

Resolutions

R. H. Carpenter, New York, was introduced to the meeting and offered the following resolutions:

RESOLVED, that our sincere thanks, appreciation and continued interest be and hereby are, extended:

To the *Illinois Chapter,* our gracious host, who through its able and efficient committees has shown its boundless hospitality so well, and has done so much to make this meeting thoroughly enjoyable from every standpoint.

To the *City of Chicago* and its capable and genial Mayor Kelly, who gave us such a hearty and kindly welcome.

To the *Association of Commerce,* who stood so solidly behind our local chapter in urging us to meet here, and for its assistance in making our stay so pleasant.

To the *Management of the Palmer House,* which has once more demonstrated its ability to look after our every need in making us comfortable.

To the *Press,* both trade and daily, for the fine amount of publicity given the Society and its activities both before and during this meeting.

To all the *Exhibitors* of the Fourth International Heating and Ventilating Exposition, and to C. F. Roth, the exposition manager, for their helpful cooperation in coordinating their activities with our own during this meeting.

To the *Saddle and Sirloin Club* for its thoughtful courtesy in entertaining our Council so royally during its Monday meeting, and also for the extending of guest privileges to us.

To the *American Rolling Mill Company* for its radio tribute to the air conditioning industry during their nationwide broadcast last evening, and

To the *Railroads and City Transportation Companies* for their helpful and efficient handling of transportation arrangements for members and guests.

R. H. CARPENTER,
J. D. CASSELL,
W. T. JONES,

COMMITTEE ON RESOLUTIONS.

The meeting adjourned at 12:20 P.M.

PROGRAM 42ND ANNUAL MEETING

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS
PALMER HOUSE, CHICAGO, ILL.
JANUARY 27-30, 1936

TECHNICAL SESSIONS

Monday, January 27

10:00 A.M. Business Meeting—Red Lacquer Room
Call to Order—Pres. John Howatt
Reports of Officers of the Society
Reports of Council Committees
Reports of Special Committees—
Committee on Research, Prof. A. P. Kratz, *Chairman*
Guide Publication Committee, Prof. G. L. Larson, *Chairman*
Committee on Ventilation Standards, W. H. Driscoll, *Chairman*
Committee on Constitution and By-Laws, W. T. Jones, *Chairman*
Discussion of Proposed Standard of Rating and Testing Air Conditioning Equipment, W. E. Stark, *Chairman*
Report of Tellers of Election, R. H. Carpenter, *Chairman*

3:00 P.M. Meeting of the Council

Tuesday, January 28

9:30 A.M. Technical Session—Red Lacquer Room
Greetings—J. H. Milliken, President, Illinois Chapter
Response—President John Howatt
Welcome to Chicago—Hon. Edw. J. Kelly, Mayor of Chicago
Thermal Properties of Concrete Construction—F. B. Rowley, A. B. Algren and Clifford Carlson
Comparative Study of Combustion Results with Various Thermostats—B. E. Shaw
Fuel Saving Resulting from the Use of Storm Windows and Doors—A. P. Kratz and S. Konzo

2:30 P.M. Meeting of Committee on Research
2:30 P.M. Meeting of Chapter Officers, F. C. McIntosh, *Chairman*
4:00 P.M. Organization Meeting of Nominating Committee

Wednesday, January 29

9:30 A.M. Technical Session—Red Lacquer Room
Performance of Fin-Tube Units for Air Heating, Cooling and Dehumidifying—G. L. Tuve
Subjective Reactions of Human Beings to Certain Outdoor Atmospheric Conditions—C.-E. A. Winslow and L. P. Herrington
Ventilation Requirements—C. P. Yaglou, E. C. Riley and D. I. Coggins
Airfoil Fan Characteristics—W. A. Rowe

2:00 P.M. Warm Air Heating and Cooling Session with National Warm Air Heating and Air Conditioning Association at Stevens Hotel
Room Surface Temperature of Glass Windows—J. E. Emswiler and W. C. Randall
Study of Summer Cooling in the Research Residence Using Water from the City Water Mains—A. P. Kratz, M. K. Fahnestock, S. Konzo and E. L. Broderick

Thursday, January 30

9:30 A.M. Technical Session—Red Lacquer Room
Comfort Standards for Summer Air Conditioning—F. C. Houghten and Carl Gutberlet
Unfinished Business
Installation of Officers
New Business
Resolutions
Adjournment

12:30 P.M. Luncheon and Meeting of the Council
2:30 P.M. Meeting of Guide Publication Committee

ENTERTAINMENT EVENTS

Monday, January 27

9:30 A.M. Reception and Registration of Members, Ladies and Guests—Foyer of Red Lacquer Room
12:30 P.M. Luncheon Council—Saddle and Sirloin Club, Stock Yards Inn
2:00 P.M. Opening of 4th International Heating and Ventilating Exposition, International Amphitheatre, 42nd and Halsted Sts. (Gray Line Bus Service from Palmer House at frequent intervals from 10:37 A.M. to 8:37 P.M.—fare 15c.)
4:00 P.M. Informal tea for Ladies in Illinois Chapter Convention Room
8:30 P.M. Informal Reception for members and ladies in Illinois Chapter Convention Room, Palmer House, featuring Miss Alice Blue, famous pianist of Station WGN

Tuesday, January 28

10:30 A.M. Chicago Art Institute Tour for Ladies—Assemble in Illinois Chapter Convention Room
12:30 P.M. Get-Acquainted Luncheon for members and guests, Chicago Room, Palmer House
1:00 P.M. Ladies' Luncheon and Style Show at Marshall Field & Co. (Assemble at Illinois Chapter Convention Room at 12:45 P.M.)

12:00 to 10:30 P.M. 4th International Heating and Ventilating Exposition at International Amphitheatre. Tickets of admission available at Registration Headquarters

7:00 P.M. Past-Presidents Dinner, Palmer House

10:00 P.M. House-Warming Party—Red Lacquer Room, Palmer House—Dancing and Entertainment—Admission tickets at Registration Headquarters

Wednesday, January 29

10:45 A.M. Tour of Adler Planetarium for Ladies

12:00 to 10:30 P.M. 4th International Heating and Ventilating Exposition at International Amphitheatre

7:00 P.M. Annual Banquet and Dance—Grand Ballroom—Palmer House, Broadcast by Pres. John Howatt over N.B.C. Blue Network. . . . Presentation of F. Paul Anderson Medal to Dr. Arthur Cutts Willard, President, University of Illinois by Pres. John Howatt. . . . Presentation of past president's emblem to Mr. Howatt by Samuel R. Lewis. . . . Speaker—Major John L. Griffith, Competition in Athletics and in Business. . . . Toastmaster, Oliver J. Prentice. . . . Entertainment by Original Old Heidelburg Octette. . . . Music for Dancing by Waddy Wadsworth and his Band.

Thursday, January 30

12:00 to 10:30 P.M. 4th International Heating and Ventilating Exposition, International Amphitheatre

2:00 P.M. Inspection of Manufacturing Plants—register your choice for these trips at Registration Headquarters

Friday, January 31

10:00 A.M. Inspection of Thermal Engineering Laboratories, test rooms, foundries, forge, finishing and pipe shops of Crane Co., Chicago (Corwith) Plant. . . . Details at Registration Headquarters

COMMITTEE ON ARRANGEMENTS

J. H. Milliken, *Chairman* J. J. Hayes, *Vice-Chairman*

Publicity—C. E. Price, *Chairman;* Geo. G. Turner, L. W. Britton.

Reception—J. R. Vernon, *Chairman;* H. P. Reid, O. W. Armspach, Eugene Mathis, J. J. Haines, C. W. DeLand, R. E. Hattis, A. B. Martin, J. C. Matchett, Homer R. Linn, J. M. Stannard.

Ladies—C. F. Hayden, *Chairman;* Mrs. Byron L. Casey, Mrs. C. W. DeLand, Mrs. J. P. Fleming, Mrs. T. F. Hanley, Mrs. C. F. Hayden, Mrs. J. J. Hayes, Mrs. E. P. Heckel, Mrs. F. P. Keeney, Mrs. J. C. Matchett, Mrs. J. E. McClellan, Mrs. E. N. McDonnell, Mrs. J. H. Milliken, Mrs. R. E. Moore, Mrs. L. L. Narowetz, Jr., Mrs. C. E. Price, Mrs. F. I. Raymond, Mrs. Lester Ries, Mrs. S. I. Rottmayer, Mrs. A. G. Sutcliffe, Mrs. G. G. Turner, Mrs. J. R. Vernon.

Transportation—Byron L. Casey, *Chairman;* E. M. Mittendorff, F. C. Warren.

Entertainment—Tom Brown, *Chairman;* W. G. Jennings, E. T. Murphy, H. G. Kreissl, E. N. McDonnell, L. W. Washington, L. L. Narowetz, Jr., C. E. Crone, Jr.

Banquet—J. P. Fleming, *Chairman;* John J. Aeberly, John R. Borling, R. B. Hayward, A. E. Christophersen, Thos. F. Hanley, Jr., Jas. J. Finan, Roy M. Getschow, H. B. Hinckley.

Finance—Elmer W. Rietz, *Chairman;* S. M. Trumbo, E. J. Gossett.

Advisory Committee—John F. Hale, Dr. E. V. Hill, H. M. Hart, Samuel R. Lewis.

No. 1026

THERMAL PROPERTIES OF CONCRETE CONSTRUCTION

By F. B. ROWLEY,* A. B. ALGREN ** (*MEMBERS*), AND CLIFFORD CARLSON *** (*NON-MEMBER*), MINNEAPOLIS, MINN.

This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, in cooperation with the *Portland Cement Association* and the University of Minnesota

THERE is no doubt about the necessity of building high thermal resistance into the outside walls of many of our buildings. It would be a comparatively simple job to get high thermal resistance if this were the only quality required, but most buildings have other requirements which make it necessary to select materials and types of construction which have economic structural properties as well as high resistance. It is usually true that materials with high structural strength have low heat resistance, making it necessary to give special consideration to the methods of using them, and perhaps to add special insulation where thermal resistance is an important factor. Concrete is such a material. It has excellent structural properties, but when the ordinary aggregates are used to build monolithic walls without some consideration as to insulating values, a prohibitive thermal conductivity may result.

The thermal properties of concrete walls may be improved by such methods as changing the nature of the aggregate, constructing the wall with different types of air spaces, applying a special surface finish or adding some insulating material to the wall. The purpose of this investigation was to study the different types of masonry and monolithic concrete construction and to find what practical method might be employed to increase thermal resistance. The investigation has thus far included monolithic and masonry walls, different aggregates, different surface finishes, various types of air space construction and the application of specific insulating materials. Obviously there are many combinations which might be considered, but in the interest of economy the number of walls actually built and tested was limited to representative types.

Aggregates

In general low density aggregates will result in lower thermal conductivity than will high density aggregates. This, however, is not always the case as low

* Director of Experimental Engineering Laboratories, University of Minnesota.
** Assistant Professor of Mechanical Engineering, University of Minnesota.
*** Research Engr., University of Minnesota.

Presented at the 42nd Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Chicago, Ill., January, 1936, by F. B. Rowley.

density may be accompanied by an excessive porosity which will allow the heat to be transferred through the wall by air circulation. Such a condition may be taken care of in part by the application of some impervious surface finish. This will exclude the surface air, but in extreme cases there may also be circulation of air within the wall itself. In this investigation sand and crushed limestone, sand and gravel, cinders and Haydite were used as aggregates. These were graded as indicated by the description of the individual walls.

Air Space Construction Used

There are two types of air spaces which may be built into a wall. First, the continuous air space as may be provided between two parallel walls; and second, irregular cored-out sections as are usually provided in concrete block construction. In the first case all heat conducted from surface to surface must pass directly across the air space, and, if there are no connecting rods which serve as conductors, the efficiency of the air space could be very definitely determined from the known properties of such spaces. In the second case only a part of the heat must pass through the air spaces, the remainder passing through the solid material between the spaces. If the material proper has a high thermal conductivity and if the partitions between the cored-out spaces are direct between the two surfaces and constitute a considerable percentage of the wall area, the effectiveness of the air space as an insulator may be very materially reduced. It is difficult to make calculations to determine the thermal conductivity of walls containing irregular air spaces, even though the properties of the material are fairly well established. An efficient air space construction must either provide a continuous space over the full wall area or, if there are paths of solid material between the two surfaces, they should be designed as long and restricted in area as possible. In this investigation parallel or continuous air spaces were provided by constructing walls of parallel monolithic and masonry slabs and by furring out the surface for specific insulating finish. The cored air spaces were those commonly used in standard concrete block construction.

Surface Finish

The effect of surface finish on a wall may be to reduce the normal air filtration into and out of the wall and thus improve the internal conductivity by eliminating convection currents, or to change the character of the surface and thereby change the surface coefficient of conductance. If the surface finish should be of material thickness and of an insulating value in itself, it would add to the thermal resistance of the wall. The surface finishes used in this investigation were for the purpose of reducing any interchange of air through the boundary surfaces rather than for improving the surface conductance.

Insulation

There are several methods by which thermal resistance may be built into a wall. As discussed previously, air space construction may be used for this purpose or specific insulating materials may be applied to the walls either in air spaces or on the surface. The effectiveness of any application will depend upon the amount of heat which must normally flow through the insulation. Thus, if insulating material is placed over the surface of the wall in a layer of

uniform thickness, all heat must presumably pass through the material and it may be considered as an effective application. If the insulating material is placed in such manner as not to obstruct all paths for the flow of heat its effectiveness may be reduced. In this investigation insulating materials have been used in the cells or cores of the concrete blocks and also as a uniform sheet parallel to the surfaces of the walls.

METHOD OF PROCEDURE

In all cases the aggregate was selected and graded as per specifications. In each case a sieve analysis was made and the different aggregates were mixed in the required proportion to give the proper fineness modulus. Preliminary tests were made to determine the proper water to cement ratio to give the required slump and strength characteristics. The blocks for the masonry walls were constructed in a standard block machine at the Crown Sidewalk and Block Co.'s plant, Minneapolis, Minn. These were steam cured for 24 hours and then exposed to the air for the remainder of the curing period. After being thoroughly cured and dried out they were built into test walls 5½ ft square with standard ⅜ in. mortar joints between the blocks. In constructing the walls the mortar was buttered on the edges but not on the cross partitions between the air cells, thus following recommended and standard construction practice for walls of this type. After a sufficient curing period to thoroughly dry out the mortar between the joints the walls were tested by the standard hot box method.

For the monolithic walls the aggregates were selected and graded in the same manner as for the concrete blocks and the walls were constructed in the laboratory according to specifications. At the time of building the walls 6 in. x 12 in. cylinders were cast which were later tested to give the strength of the concrete. After the walls were thoroughly cured and dried out they were tested by the standard hot box method to determine their thermal conductivity.

DESCRIPTION OF WALLS TESTED

The complete physical data covering walls tested are shown in Tables 1 and 2 for masonry and monolithic walls, respectively. The cinders used in this construction were obtained from the Northern States Power Riverside Plant, Minneapolis, Minn.

The construction data for the walls are shown in Tables 3 and 4, for the masonry and monolithic walls, respectively. Figs. 1, 2 and 3 show the detailed dimensions of the different blocks used. The mortar used in building the masonry walls consisted of one part cement to three parts plaster sand, and to this mixture was added 10 per cent lime.

Walls No. 8a and 8b were built up of two 4-in. cinder tile walls with tie rods and separated by a 1-in. air space. The tie rods were made of 28-gage galvanized iron about ¾ in. wide and of sufficient length so that they extended approximately 3 in. in each wall and across the air space. They were placed on top of each course of tile 12 in. on center horizontally and grouted in place with mortar.

Walls No. 35a, 35b, and 35c were made of two 4-in. sand and gravel dry tamp monolithic walls spaced 2½ in. apart by means of tie rods. The tie rods

TABLE 1. MASONRY WALLS—

Wall No.	Type of Aggregate	Description of Block	Density of Material in Blocks Lb/Cu Ft	Percentage of Core Volume	Overall Thickness of Wall in Inches	Inside Surface Finish	Outside Surface Finish	Additional Insulation in Wall	Density of Insulating Material Lb/Cu Ft	Weight of Insulating Material per Sq Ft of Wall Area Exclusive of Surface Finish
1a	Cinder Block	8 in. x 8 in. x 16 in. 3-Oval Core	86.2	39.8	7.88	As Laid	As Laid	None	.....	.....
1b	Cinder Block	8 in. x 8 in. x 16 in. 3-Oval Core	86.2	39.8	7.88	As Laid	Two Coats Waterproofed White Portland Cement Paint	None	.....	.....
1c	Cinder Block	8 in. x 8 in. x 16 in. 3-Oval Core	86.2	39.8	7.88	As Laid	Two Coats Waterproofed White Portland Cement Paint	Cores Filled with Granulated Cork	5.12	1.34
1d	Cinder Block	8 in. x 8 in. x 16 in. 3-Oval Core	86.2	39.8	7.88	As Laid	Two Coats Waterproofed White Portland Cement Paint	Cores Filled with Dry Cinders	69.7	18.29
1e	Cinder Block	8 in. x 8 in. x 16 in. 3-Oval Core	86.2	39.8	7.88	As Laid	Two Coats Waterproofed White Portland Cement Paint	Cores Filled with Rock Wool	14.21	3.73
1f	Cinder Block	8 in. x 8 in. x 16 in. 3-Oval Core	86.2	39.8	9.38	½ in. Plaster on Metal Lath Furred 1 in.	Two Coats Waterproofed White Portland Cement Paint	Cores Filled with Rock Wool	14.21	3.73
1g	Cinder Block	8 in. x 8 in. x 16 in. 3-Oval Core	86.2	39.8	8.38	½ in. Plaster Applied Direct	As Laid	Cores Filled with Rock Wool	14.21	3.73
1h	Cinder Block	8 in. x 8 in. x 16 in. 3-Oval Core	86.2	39.8	9.88	½ in. Insulation Board Furred 1 in.	½ in. Plaster Applied Direct	Cores Filled with Rock Wool	14.21	3.73
2a	Haydite Block	8 in. x 8 in. x 16 in. 3-Oval Core	67.7	39.6	7.91	As Laid	As Laid	None	.....	.....
2b	Haydite Block	8 in. x 8 in. x 16 in. 3-Oval Core	67.7	39.6	7.91	As Laid	Two Coats Waterproofed White Portland Cement Paint	None	.....	.....

DESCRIPTION AND DATA

Aggregate Used in Wall Per Cent by Weight	Fineness Modulus of Aggregate			Dry Rodded Weight of Aggregate Lb per Cu Ft		Mix Proportion Dry Rodded Volume	Water-Cement Ratio W/C Gal/Sack	28-Day Compressive Strength Air Dried					Per Cent Absorption at 28 Days	
								Area of Specimen Sq In.		Total Breaking Load Lb	Breaking Load Lb/Sq In.			
	Fine	Coarse	Combined	Fine	Coarse			Gro	Net		Gro	Net	By Wt	By Vol
100 Per Cent Cinders ½ in. to 0	4.11	.....	4.11	65	....	1:8	8.75	124.9	75.13	99,263	795	1321	14.2	19.7
100 Per Cent Cinders ½ in. to 0	4.11	.....	4.11	65	....	1:8	8.75	124.9	75.13	99,263	795	1321	14.2	19.7
100 Per Cent Cinders ½ in. to 0	4.11	.....	4.11	65	....	1:8	8.75	124.9	75.13	99,263	795	1321	14.2	19.7
100 Per Cent Cinders ½ in. to 0	4.11	.....	4.11	65	....	1:8	8.75	124.9	75.13	99,263	795	1321	14.2	19.7
100 Per Cent Cinders ½ in. to 0	4.11	.....	4.11	65	....	1:8	8.75	124.9	75.13	99,263	795	1321	14.2	19.7
100 Per Cent Cinders ½ in. to 0	4.11	.....	4.11	65	....	1:8	8.75	124.9	75.13	99,263	795	1321	14.2	19.7
100 Per Cent Cinders ½ in. to 0	4.11	.....	4.11	65	....	1:8	8.75	124.9	75.13	99,263	795	1321	14.2	19.7
100 Per Cent Cinders ½ in. to 0	4.11	.....	4.11	65	....	1:8	8.75	124.9	75.13	99,263	795	1321	14.2	19.7
100 Per Cent Haydite 64 Per Cent Size A	2.76	5.94	3.92	58	43	1:8½	10.82	125.5	75.75	97,780	779	1289	24.4	25.5
36 Per Cent Size B by Weight	2.76	5.94	3.92	58	43	1:8½	10.82	125.5	75.75	97,780	779	1289	24.4	25.5

TABLE 1. MASONRY WALLS—

Wall No.	Type of Aggregate	Description of Block	Density of Material in Blocks Lb/Cu Ft	Percentage of Core Volume	Overall Thickness of Wall in Inches	Inside Surface Finish	Outside Surface Finish	Additional Insulation in Wall	Density of Insulating Material Lb/Cu Ft	Weight of Insulating Material per Sq Ft of Wall Area Exclusive of Surface Finish
3a	Sand and Gravel Block	8 in. x 8 in. x 16 in. 3-Oval Core	126.4	39.8	7.88	As Laid	As Laid	None	.....	.....
3b	Sand and Gravel Block	8 in. x 8 in. x 16 in. 3-Oval Core	126.4	39.8	7.88	As Laid	Two Coats Waterproofed White Portland Cement Paint	None	.....	.....
4a	Limestone Block	8 in. x 8 in. x 16 in. 3-Oval Core	134.3	40.2	7.83	As Laid	As Laid	None	.....	.....
5a	Cinder Block	8 in. x 12 in. x 16 in. 3-Oval Core	86.2	40.2	11.88	As Laid	As Laid	None	.....	.....
6a	Sand and Gravel Block	8 in. x 12 in. x 16 in. 3-Oval Core	124.9	40.4	11.84	As Laid	As Laid	None	.....	.....
7a	Cinder Tile	4 in. x 8 in. x 16 in. 3-Core Partition	99.9	33.9	4.17	As Laid	As Laid	None	.....	.....
8a	Cinder Tile	4 in. x 8 in. x 16 in. 3-Core Partition	99.9	33.9	9.33	As Laid	As Laid	1 in. Air Space between Partition Tile	.....	.....
8b	Cinder Tile	4 in. x 8 in. x 16 in. 3-Core Partition	99.9	33.9	9.33	As Laid	As Laid	1 in. Rock Wool between Partition Tile	9.97	0.830

were ¼ in. in diameter and extended about 3 in. in each wall. They were spaced 12 in. on centers horizontally and 9 in. on centers vertically in the test section.

TEST RESULTS

The complete thermal conductivity data for the walls, obtained by the hot box method, are given in Tables 5 and 6. These tables also include some of the physical characteristics of the walls taken from Tables 1 and 2 in order to more easily identify the wall construction when referring to the thermal properties.

DESCRIPTION AND DATA—(*Continued*)

Aggregate Used in Wall Per Cent by Weight	Fineness Modulus of Aggregate			Dry Rodded Weight of Aggregate Lb per Cu Ft		Mix Proportion Dry Rodded Volume	Water-Cement Ratio W/C Gal/Sack	28-Day Compressive Strength Air Dried					Per Cent Absorption at 28 Days	
								Area of Specimen Sq In.		Total Breaking Load Lb	Breaking Load Lb/Sq In.			
	Fine	Coarse	Combined	Fine	Coarse			Gro	Net		Gro	Net	By Wt	By Vol
73 Per Cent Sand	3.22	5.68	3.91	108	105	1:10	8.92	124.9	75.18	137,440	1100	1829	8.3	16.7
27 Per Cent Pea Size Gravel by Weight	3.22	5.68	3.91	108	105	1:10	8.92	124.9	75.18	137,440	1100	1829	8.3	16.7
78 Per Cent Limestone Screenings 22 Per Cent Pea Size Limestone by Weight	3.33	5.89	4.00	111	94	1:9	9.92	123.6	73.88	198,050	1602	2683	7.9	17.0
100 Per Cent Cinders ½ in. to 0	4.05	.....	4.05	63	....	1:8	12.30	188.5	112.8	154,557	820	1369	14.4	19.9
73 Per Cent Sand 27 Per Cent Pea Size Gravel by Weight	3.22	5.68	3.91	108	105	1:10	6.82	187.6	111.9	169,340	903	1512	7.32	14.7
100 Per Cent Cinders ½ in. to 0	4.11	.....	4.11	65	....	1:8	7.72	66.3	43.8	55,600	838	1268	11.95	19.1
100 Per Cent Cinders ½ in. to 0	4.11	.....	4.11	65	....	1:8	7.72	66.3	43.8	55,600	838	1268	11.95	19.1
100 Per Cent Cinders ½ in. to 0	4.11	.....	4.11	65	....	1:8	7.72	66.3	43.8	55,600	838	1268	11.95	19.1

In taking test data there was some difficulty in getting consistent surface temperature from which to calculate surface conductance coefficients. This was due to the fact that the surfaces of several of the walls were somewhat rough, making it impossible to get precision measurements of surface temperatures, and also because many of the walls contained irregular cored-out spaces which gave different heat resistances at different sections of the wall. For example, the surface temperatures for a hollow concrete block wall would be different when taken at a point opposite an air space than when taken at a point opposite the partition between air spaces. In some of the ribbed wall construction as much as 5 deg was noted between those points over the ribs as

TABLE 2. MONOLITHIC WALLS—

Wall No.	Type of Aggregate	Description of Wall	Density of Concrete in Wall Lb/Cu Ft	Overall Thickness of Wall in Inches	Inside Surface Finish	Outside Surface Finish	Additional Insulation	Density of Insulating Material Lb/Cu Ft	Weight of Insulation per Sq Ft of Wall Area
30a	Sand and Limestone	Sand and Limestone —Plastic Mix	140.3	4.32	As Cast	As Cast	None	......	......
31a	Sand and Gravel	Sand and Coarse Gravel — Plastic Mix	143.3	4.19	As Cast	As Cast	None	......	......
32a	Sand and Gravel	Sand and Coarse Gravel — Dry Tamp Mix	148.8	4.05	As Cast	As Cast	None	......	......
33a	Cinder	Cinder Aggregate Plastic Mix	94.4	3.90	As Cast	As Cast	None	......	......
34a	Haydite	Haydite Aggregate Plastic Mix	77.7	3.96	As Cast	As Cast	None	......	......
35a	Sand and Gravel	Two Sand and Gravel Slabs — Dry Tamped 2½ in. Air Space	148.25	10.66	As Cast	As Cast	2.52 in. Air Space	......	......
35b	Sand and Gravel	Two Sand and Gravel Slabs — Dry Tamped 2½ in. Air Space	148.25	10.66	As Cast	As Cast	2.52 in. Dry Cinders	75.4	15.87
35c	Sand and Gravel	Two Sand and Gravel Slabs — Dry Tamped 2½ in. Air Space	148.25	10.66	As Cast	As Cast	2.52 in. Rock Wool	14.3	2.99
36a	Cinder	Sand and Coarse Cinders — Dry Tamped	118.7	3.96	As Cast	As Cast	None	......	......
37a	Sand and Gravel	Sand and Gravel— Plastic Mix 2 in. Slab with 4 in. x 4 in. Ribs 24 in. O. C.	143.2	2.18	As Cast	As Cast	None	......	......
37b	Sand and Gravel	Sand and Gravel.— Plastic Mix 2 in. Slab with 4 in. x 4 in. Ribs 24 in. O. C. 5 in. Air Space	143.2	7.68	½ in. Plaster on Metal Lath Furred 1 in. on Ribs	As Cast	5 in. Air Space	......	......
37c	Sand and Gravel	Sand and Gravel Plastic Mix 2 in. Slab with 4 in. x 4 in. Ribs 24 in. O. C. 5 in. Rock Wool	143.2	7.68	½ in. Plaster on Metal Lath Furred 1 in. on Ribs	As Cast	5 in. Rock Wool	9.53	3.63

DESCRIPTION AND DATA

Aggregate Used in Wall Per Cent by Weight	Fineness Modulus of Aggregate			Dry Rodded Weight of Aggregate Lb per Cu Ft		Mix Proportion Dry Rodded Volume	Water Cement Ratio W/C Gal per Sack	Average Slump Inches	28-Day Compressive Strength Wet			Per Cent Absorption at 28 Days	
	Fine	Coarse	Combined	Fine	Coarse				Area of Cylinder Sq In.	Total Breaking Load Lb	Breaking Load in Lb/Sq In.	By Wgt	By Vol
39.3 Per Cent Sand 60.7 Per Cent Crushed Limestone	3.00	6.80	5.33	109.9	97.53	1:2¼:4	8.0	5.50	28.84	111,253	3,853	6.02	13.51
36.5 Per Cent Sand 63.5 Per Cent Gravel	3.00	6.71	5.36	109.9	107.6	1:2¼:4	8.0	5.90	28.59	107,867	3,772	5.24	12.03
36.5 Per Cent Sand 63.5 Per Cent Gravel	3.00	6.71	5.36	109.9	107.6	1:2¼:4	6.0	0.50	28.25	152,173	5,385	4.17	9.95
100 Per Cent Cinders 39.8 Per Cent Fine 60.2 Per Cent Coarse	3.25	6.77	5.37	65.5	55.64	1:2¼:4	11.2	3.94	28.06	28,312	1,009	16.55	24.98
100 Per Cent Haydite 42.6 Per Cent Fine 57.4 Per Cent Coarse	2.76	6.67	5.00	58.6	44.41	1:2¼:4	10.5	4.87	28.8	64,992	2,252	23.07	28.80
36.5 Per Cent Sand 63.5 Per Cent Gravel	3.00	6.71	5.36	109.9	107.6	1:2¼:4	6.0	0.30	28.21	133,366	4,728	4.34	10.31
36.5 Per Cent Sand 63.5 Per Cent Gravel	3.00	6.71	5.36	109.9	107.6	1:2¼:4	6.0	0.30	28.21	133,366	4,728	4.34	10.31
36.5 Per Cent Sand 63.5 Per Cent Gravel	3.00	6.71	5.36	109.9	107.6	1:2¼:4	6.0	0.30	28.21	133,366	4,728	4.34	10.31
52.6 Per Cent Sand 47.4 Per Cent Cinders	3.00	6.77	5.21	109.9	55.64	1:2¼:4	6.75	0.30	28.2	90,850	3,218	8.35	15.67
36.5 Per Cent Sand 63.5 Per Cent Gravel	3.00	6.71	5.36	109.9	106.5	1:2¼:4	8.68	7.49	28.57	78,657	2,857	6.15	14.42
36.5 Per cent Sand 63.5 Per cent Gravel	3.00	6.71	5.36	109.9	106.5	1:2¼:4	8.68	7.49	28.57	78,657	2,857	6.15	14.42
36.5 Per cent Sand 63.5 Per cent Gravel	3.00	6.71	5.36	109.9	106.5	1:2¼:4	8.68	7.49	28.57	78,657	2,857	6.15	14.42

TABLE 3. CONSTRUCTION DATA FOR MASONRY WALLS

WALL No.	TYPE OF AGGREGATE IN BLOCKS	AGGREGATE GRADING AND AVERAGE FINENESS MODULUS	BLOCKS MANUFACTURED DATE	WALLS BUILT DATE	INSIDE SURFACE TREATMENT	DATE OF TREATMENT	OUTSIDE SURFACE TREATMENT	DATE OF TREATMENT
1a	Cinders		11-23-34	1-3-35	None		None	
1b	Cinders		11-23-34	1-3-35	None		2 Coats Water Proofed White Portland Cement Paint	3-13-35
1c	Cinders		11-23-34	1-3-35	None		2 Coats Water Proofed White Portland Cement Paint	3-13-35
1d	Cinders	Average Fineness Modulus = 4.11 Per Cent Passing ½ in. — 100.0 No. 16 — 32.9 ⅜ in. — 94.4 No. 30 — 21.6 No. 4 — 71.7 No. 50 — 12.9 No. 8 — 49.3 No. 100 — 6.1	11-23-34	1-3-35	None		2 Coats Water Proofed White Portland Cement Paint	3-13-35
1e	Cinders		11-23-34	1-3-35	None		2 Coats Water Proofed White Portland Cement Paint	3-13-35
1f	Cinders		11-23-34	1-3-35	½ in. Plaster on Metal Lath Furred 1 in.	4-9-35	2 Coats Water Proofed White Portland Cement Paint	3-13-35
1g	Cinders		11-23-34	1-3-35	½ in. Plaster Applied Direct	5-2-35 5-4-35	None	
1h	Cinders		11-23-34	1-3-35	½ in. Insulation Board Furred 1 in. Strips 16 in. on Center	5-9-35	½ in. Plaster Applied Direct	5-2-35 5-4-35

TABLE 3 (*Continued*)

2a	Haydite	Average Fineness Modulus = 3.92 Per Cent Passing ½ in. — 100.0 No. 16 — 38.2 ⅜ in. — 93.1 No. 30 — 21.9 No. 4 — 68.5 No. 50 — 14.4 No. 8 — 61.7 No. 100 — 10.0	11–23–34	1–3–35	None		None	
2b	Haydite		11–23–34	1–3–35	None		2 Coats Water Proofed White Portland Cement Paint	3–16–35
3a	Sand and Gravel	Average Fineness Modulus = 3.91 Per Cent Passing ⅜ in. — 100.0 No. 16 — 41.1 No. 4 — 79.4 No. 30 — 22.0 No. 8 — 59.5 No. 50 — 5.7 No. 100 — 1.3	11–14–34	1–3–35	None		None	
3b	Sand and Gravel		11–14–34	1–3–35	None		2 Coats Water Proofed White Portland Cement Paint	3–9–35
4a	Limestone	Average Fineness Modulus = 4.0 Per Cent Passing ⅜ in. — 100.0 No. 50 — 14.6 No. 4 — 73.7 No. 100 — 9.1 No. 16 — 32.3	11–14–34	1–2–35	None		None	
5a	Cinders	Average Fineness Modulus = 4.05 Per Cent Passing ½ in. — 100.0 No. 16 — 36.8 ⅜ in. — 90.4 No. 50 — 18.2 No. 4 — 65.1 No. 100 — 9.5	11–6–35	1–2–35	None		None	
6a	Sand and Gravel	Average Fineness Modulus = 3.91 Per Cent Passing ⅜ in. — 100.0 No. 16 — 41.1 No. 4 — 79.4 No. 30 — 22.0 No. 8 — 59.5 No. 50 — 5.7 No. 100 — 1.3	11–6–35	1–2–35	None		None	
7a	Cinders	Average Fineness Modulus = 4.11 Per Cent Passing ½ in. — 100.0 No. 16 — 32.9 ⅜ in. — 94.4 No. 30 — 21.6 No. 4 — 71.7 No. 50 — 12.9 No. 8 — 49.3 No. 100 — 6.1	11–24–35	1–4–35	None		None	
8a	Cinders		11–24–35	1–4–35	None		None	
8b	Cinders		11–24–35	1–4–35	None		None	

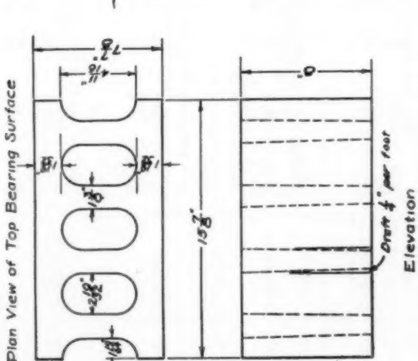

FIG. 1. 3 CORE BLOCK 8 IN. x 8 IN. x 16 IN.

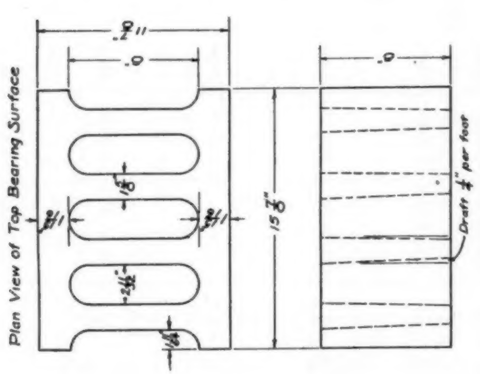

FIG. 2. 3 CORE BLOCK 8 IN. x 12 IN. x 16 IN.

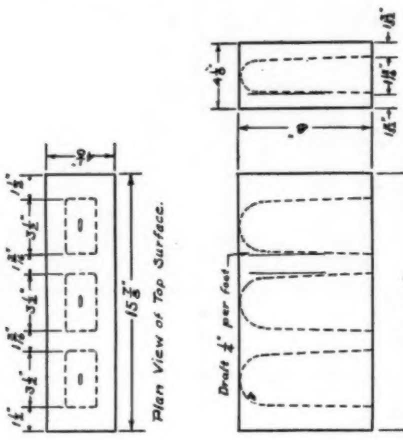

FIG. 3. 3 PARTITION CORE 4 IN. x 8 IN. x 16 IN.

TABLE 4. CONSTRUCTION DATA FOR MONOLITHIC WALLS

WALL No.	TYPE OF AGGREGATE IN WALL	GRADING OF AGGREGATE AND AVERAGE FINENESS MODULUS	MIX PROPORTIONS BY VOLUME AND KIND OF MIX	NUMBER OF LIFTS REQUIRED FOR POURING WALL	DATE OF BUILDING WALL
30a	Sand and Limestone Concrete	Average Fineness Modulus = 5.33 Per Cent Passing FINE: 3/8 in. — 100.0; No. 4 — 99.6; No. 8 — 88.8; No. 16 — 64.1; No. 30 — 36.1; No. 50 — 9.2; No. 100 — 1.8 COARSE: 1 in. — 100.0; 3/4 in. — 79.8; 1/2 in. — 64.6; 3/8 in. — 37.4; No. 4 — 3.0; No. 8 — 0.0	1:2¼:4 Plastic Mix	4	12-19-34
31a	Sand and Coarse Gravel Concrete	Average Fineness Modulus = 5.36 Per Cent Passing FINE: 3/8 in. — 100.0; No. 4 — 99.6; No. 8 — 88.8; No. 16 — 64.1; No. 30 — 36.1; No. 50 — 9.2; No. 100 — 1.8 COARSE: 1 in. — 100.0; 3/4 in. — 84.9; 1/2 in. — 65.9; 3/8 in. — 41.1; No. 4 — 3.2; No. 8 — 0.0	1:2¼:4 Plastic Mix	4	12-27-34
32a	Sand and Coarse Gravel Concrete		1:2¼:4 Dry Tamp Mix	7	1-8-35 1-9-35
33a	Cinder Concrete	Average Fineness Modulus = 5.37 Per Cent Passing FINE: No. 4 — 99.9; No. 8 — 73.2; No. 16 — 51.5; No. 30 — 33.5; No. 50 — 13.7; No. 100 — 3.7 COARSE: 1 in. — 100.0; 3/4 in. — 82.7; 1/2 in. — 64.7; 3/8 in. — 35.8; No. 4 — 4.5; No. 8 — 0.0	1:2¼:4 Plastic Mix	4	1-7-35
34a	Haydite Concrete	Average Fineness Modulus = 5.00 Per Cent Passing FINE: No. 4 — 99.9; No. 8 — 94.8; No. 16 — 58.2; No. 30 — 33.3; No. 50 — 21.5; No. 100 — 14.7 COARSE: 3/4 in. — 100.0; 1/2 in. — 88.7; 3/8 in. — 38.8; No. 4 — 5.3; No. 8 — 0.0	1:2¼:4 Plastic Mix	4	1-11-35
35a	Sand and Gravel Concrete	Average Fineness Modulus = 5.36 Per Cent Passing FINE: 3/8 in. — 100.0; No. 4 — 99.6; No. 8 — 88.8; No. 16 — 64.1; No. 30 — 36.1; No. 50 — 9.2; No. 100 — 1.8 COARSE: 1 in. — 100.0; 3/4 in. — 84.9; 1/2 in. — 65.9; 3/8 in. — 41.1; No. 4 — 3.2; No. 8 — 0.0	1:2¼:4 Dry Tamp Mix	7	1-18-35 1-19-35
35b	Sand and Gravel Concrete		1:2¼:4 Dry Tamp Mix	7	1-18-35 1-19-35
35c	Sand and Gravel Concrete		1:2¼:4 Dry Tamp Mix	7	1-18-35 1-19-35
36a	Sand and Coarse Cinder Concrete	Average Fineness Modulus = 5.21 Per Cent Passing FINE: 3/8 in. — 100.0; No. 4 — 99.6; No. 8 — 88.8; No. 16 — 64.1; No. 30 — 36.1; No. 50 — 9.2; No. 100 — 1.8 COARSE: 1 in. — 100.0; 3/4 in. — 82.7; 1/2 in. — 64.7; 3/8 in. — 35.8; No. 4 — 4.5; No. 8 — 0.0	1:2¼:4 Dry Tamp Mix	7	1-14-35
37a	Sand and Gravel Concrete	Average Fineness Modulus = 5.36 Per Cent Passing FINE: 3/8 in. — 100.0; No. 4 — 99.6; No. 8 — 88.8; No. 16 — 64.1; No. 30 — 36.1; No. 50 — 9.2; No. 100 — 1.8 COARSE: 3/4 in. — 100.0; 1/2 in. — 70.0; 3/8 in. — 40.0; No. 4 — 0.0	1:2¼:4 Plastic Mix	7	1-28-35
37b	Sand and Gravel Concrete		1:2¼:4 Plastic Mix	7	1-28-35
37c	Sand and Gravel Concrete		1:2¼:4 Plastic Mix	7	1-28-35

TABLE 5. TEST DATA AND RESULTS FOR MASONRY WALLS

Wall No.	Test No.	Date of Test	Description of Wall	Inside Surface Finish	Outside Surface Finish	Additional Insulation in Walls	Air Temperatures F			Coefficient of Heat Transmission		
							High Side	Low Side	Mean Temp.	C	U	U Corrected to 15 mph Wind Vel.
1a	2	2-12-35	8 in. x 8 in. x 16 in. 3-Oval Core Cinder Block	As laid	As laid	None	80.81	− 0.78	40.01	0.583	0.335	0.399
1a	9	3-7-35		As laid	As laid	None	80.07	0.13	40.10	0.577	0.333	0.396
1a	10	3-12-35		As laid	As laid	None	79.98	0.18	40.08	0.577	0.333	0.396
1b	15	3-26-35		As laid	Two Coats Waterproofed White Portland Cement Paint	None	79.56	0.48	40.02	0.522	0.314	0.370
1c	18	4-2-35		As laid	Two Coats Waterproofed White Portland Cement Paint	Cores filled with Granulated Cork	80.38	− 0.43	39.98	0.238	0.189	0.201
1d	17	3-30-35		As laid	Two Coats Waterproofed White Portland Cement Paint	Cores filled with Dry Cinders	80.23	− 0.27	39.98	0.364	0.249	0.283
1e	19	4-4-35		As laid	Two Coats Waterproofed White Portland Cement Paint	Cores filled with Rock Wool	80.43	− 0.41	40.01	0.254	0.192	0.211
1f	28	4-26-35		½ in. Plaster on Metal Lath furred 1 in.	Two Coats Waterproofed White Portland Cement Paint	Cores filled with Rock Wool	80.23	− 0.32	39.96	0.198	0.158	0.171
1g	37	5-28-35		½ in. Plaster applied direct	As laid	Cores filled with Rock Wool	80.54	− 0.67	39.94	0.251	0.190	0.209
1h	36	5-26-35		½ in. Insulation Board furred 1 in. strips 16 in. on centers	½ in. Plaster applied direct	Cores filled with Rock Wool	80.64	− 0.64	40.00	0.164	0.136	0.147

TABLE 5. (*Continued*)

2a	3	2–14–35	8 in. x 8 in. x 16 in. 3-Oval Core Haydite Block	As laid	As laid	None	80.07	0.0	40.03	0.498	0.305	0.357
2a	11	3–15–35		As laid	As laid	None	79.80	0.20	40.00	0.493	0.303	0.354
2b	16	3–28–35		As laid	Two Coats Waterproofed White Portland Cement Paint	None	80.23	− 0.27	39.98	0.454	0.289	0.334
3a	1	2–8–35	8 in. x 8 in. x 16 in. 3-Oval Core Sand and Gravel Block	As laid	As laid	None	80.15	0.29	40.22	0.914	0.423	0.530
3a	8	3–5–35		As laid	As laid	None	78.40	1.74	40.07	0.882	0.416	0.519
3b	13	3–22–35		As laid	Two Coats Waterproofed White Portland Cement Paint	None	79.68	0.40	40.04	0.851	0.409	0.509
4a	6	2–29–35	8 in. x 8 in. x 16 in. 3-Oval Core Limestone Block	As laid	As laid	None	78.92	0.94	39.93	0.856	0.410	0.510
5a	7	3–3–35	8 in. x 12 in. x 16 in. 3-Oval Core Cinder Block	As laid	As laid	None	79.31	0.64	39.98	0.531	0.318	0.374
6a	12	3–19–35	8 in. x 12 in. x 16 in. 3-Oval Core Sand and Gravel Block	As laid	As laid	None	79.94	0.10	40.02	0.777	0.391	0.481
7a	14	3–24–35	4 in. x 8 in. x 16 in. 3-Core Cinder Partition Tile	As laid	As laid	None	80.51	− 0.52	40.00	1.003	0.441	0.559
8a	4	2–22–35	Double 4 in. x 8 in. x 16 in. Cinder Partition Tile spaced 1 in. apart	As laid	As laid	1 in. Air Space	79.62	0.34	39.98	0.358	0.247	0.279
8b	5	2–27–35	Double 4 in. x 8 in. x 16 in. Cinder Partition Tile spaced 1 in. apart	As laid	As laid	1 in. Rock Wool in Space between Partition Tile	79.82	0.15	39.99	0.204	0.162	0.176

TABLE 6. TEST DATA AND RESULTS FOR MONOLITHIC WALLS

Wall No.	Test No.	Date of Test	Description of Wall	Inside Surface Finish	Outside Surface Finish	Insulation	Air Temperature F			Coefficient of Heat Transmission		
							High Side	Low Side	Mean Temp.	C	U	U Corrected to 15 mph Wind Vel.
30a	20	4–6–35	4 in. Sand and Limestone Concrete Plastic Mix	As Cast	As Cast	None	79.49	0.53	40.01	2.810	0.615	0.871
31a	21	4–9–35	4 in. Sand and Coarse Gravel Concrete Plastic Mix	As Cast	As Cast	None	80.61	—0.61	40.00	2.976	0.622	0.886
31a	33	5–14–35	4 in. Sand and Coarse Gravel Concrete Plastic Mix	As Cast	As Cast	None	79.38	0.71	40.04	2.918	0.620	0.882
32a	24	4–16–35	4 in. Sand and Coarse Gravel Concrete Dry Tamp Mix	As Cast	As Cast	None	79.76	0.15	39.96	3.230	0.632	0.906
33a	22	4–11–35	4 in. Cinder Concrete Plastic Mix	As Cast	As Cast	None	79.98	—0.39	39.80	1.489	0.515	0.682
33a	23	4–13–35	4 in. Cinder Concrete Plastic Mix	As Cast	As Cast	None	79.98	0.16	40.07	1.456	0.511	0.676
34a	25	4–18–35	4 in. Haydite Concrete Plastic Mix	As Cast	As Cast	None	80.21	—0.22	40.00	0.943	0.429	0.540
35a	27	4–24–35	Double 4 in. Sand and Gravel Concrete Dry Tamp Mix	As Cast	As Cast	2.52 in. Air Space	80.13	—0.29	39.92	0.660	0.359	0.433
35b	29	5–3–35	Double 4 in. Sand and Gravel Concrete Dry Tamp Mix	As Cast	As Cast	2.52 in. Dry Cinders	80.27	—0.43	39.92	0.509	0.320	0.378
35c	34	5–17–35	Double 4 in. Sand and Gravel Concrete Dry Tamp Mix	As Cast	As Cast	2.52 in. Rock Wool	80.71	—0.78	39.97	0.170	0.140	0.150
36a	26	4–20–35	4 in. Sand and Coarse Cinder Concrete Dry Tamp Mix	As Cast	As Cast	None	80.20	—0.12	40.04	2.160	0.577	0.797
37a	35	5–21–35	2 in. Sand and Gravel Concrete with 4 in. × 4 in. Ribs, Plastic Mix	As Cast	As Cast	None	79.47	0.63	40.05	5.710	0.691	1.032
37b	32	5–10–35	2 in. Sand and Gravel Concrete with 4 in. × 4 in. Ribs, Plastic Mix	½ in. Plaster on metal lath furred 1 in. on Ribs	As Cast	5 in. Air Space	80.07	—0.18	39.95	0.687	0.367	0.445
37c	30	5–6–35	2 in. Sand and Gravel Concrete with 4 in. × 4 in. Ribs, Plastic Mix	½ in. Plaster on metal lath furred 1 in. on Ribs	As Cast	5 in. Rock Wool	80.86	—0.90	39.98	0.143	0.121	0.128
37c	31	5–8–35	2 in. Sand and Gravel Concrete with 4 in. × 4 in. Ribs, Plastic Mix	½ in. Plaster on metal lath furred 1 in. on Ribs	As Cast	5 in. Rock Wool	79.80	0.42	40.11	0.143	0.121	0.128

compared to those over the air spaces. In such cases it would be impossible to get any consistent surface coefficient without taking the average of a large number of surface temperatures. Since the overall transmission coefficients were calculated on the basis of the differences in air temperatures between the two sides they were not affected by the irregularities in surface temperatures and were, therefore, taken as a basis for computing the results and making comparisons between the various types of walls. In order to take care of the

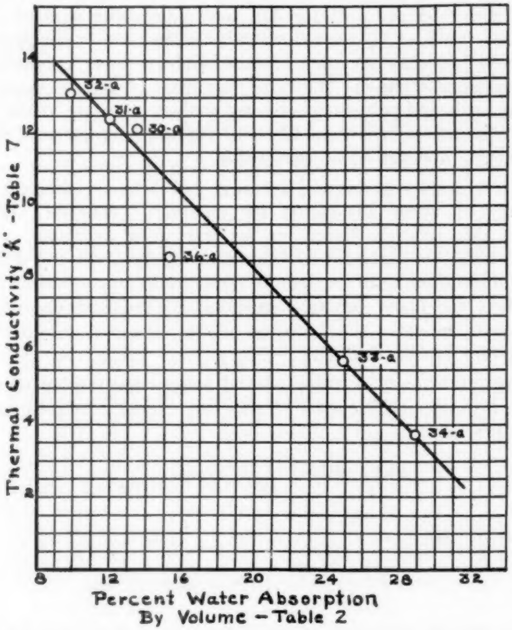

FIG. 4. CURVE SHOWING RELATION BETWEEN THERMAL CONDUCTIVITY AND WATER ABSORPTION FOR MONOLITHIC CONCRETE

irregularities in surface temperatures as mentioned, and the resulting differences in surface coefficients, averages were taken of all of the outside surface coefficients and all of the inside surface coefficients and these averages were taken as the true value for each test. From these coefficients the surface temperatures were corrected. The average of all outside surface coefficients was 1.55 and of all the inside coefficients 1.6.

Table 7 shows thermal conductivity values for the different types of concrete used in the monolithic walls.

Wall No. 35a (not shown in the Table) is a sand and gravel dry tamp mix consisting of two 4-in. monolithic slabs with a 2½-in. air space between. It is

of the same material and construction as 32a shown in the Table. Assuming an air space conductance of 1.1, the calculated conductivity value of the concrete is found to be 13. This checks with the conductivity of wall 32a which gave a test value of $k = 13.1$.

Wall 31a is of the same material as 32a, the difference being that it was made up into a plastic mix while 32a was a dry tamp mix. The thermal conductivity of 31a was 12.4 as compared with 13.1 for 32a. By referring to Table 2 it will be noted that wall 31a has an absorption of 12.03 per cent as compared with 9.95 per cent for 32a, and a breaking load of 3772 as compared with 5385 for 32a. These values indicate that 31a is slightly more porous than 32a, which probably accounts for the slightly lower conductivity. Walls 30a and 31a have substantially the same conductivity and as shown in Table 2 their breaking loads and percentages of water absorption are about equal. From these tests a conductivity of from 12 to 13 for the average sand and gravel or limestone concrete appears to be reasonable.

From the test results a relation is suggested between thermal conductivity and percentage of water absorption for monolithic concrete. The curve of Fig. 4 shows this relation, based on the tests now available, and while the investigation should be extended to more thoroughly cover the different aggregates before attempting to draw definite conclusions, the relation appears to be sufficiently close for many practical purposes. The percentage by volume of water absorption indicates the porosity of the concrete, which is one of the governing factors for conductivity.

From the results shown in Tables 7 and 8 the reduction in thermal conductivity for the 8-in. monolithic wall when using cinders, as compared with that for sand and gravel, is

$$\frac{13.1 - 5.75}{13.1} \times 100 = 56.1 \text{ per cent,}$$

and for masonry walls it is,

$$\frac{0.882 - 0.577}{0.882} \times 100 = 34.6 \text{ per cent.}$$

When using Haydite as an aggregate the reduction in thermal conductivity, as compared with sand and gravel, is

$$\frac{13.1 - 3.73}{13.1} \times 100 = 71.5 \text{ per cent for monolithic walls, and}$$

$$\frac{0.882 - 0.495}{0.882} \times 100 = 43.9 \text{ per cent for masonry walls.}$$

The ratio of the reduction for monolithic, as compared to masonry walls, is $\frac{56.1}{34.6}$ or 1.62 when using cinders, and $\frac{71.5}{43.9}$ or 1.63 when using Haydite.

The reason that the reduction is 62 per cent greater for the monolithic walls than for the block walls is due to the fact that a part of the heat passes directly through the air spaces in the masonry walls and is not affected by the kind of aggregate used.

TABLE 7. EFFECTS OF AGGREGATE ON CONDUCTIVITY FOR MONOLITHIC WALLS

WALL No.	DESCRIPTION	CONDUCTANCE C	THICKNESS INCHES	THERMAL CONDUCTIVITY k
30a	4 in. Sand and Limestone Concrete Plastic Mix	2.810	4.318	12.14
31a	4 in. Sand and Coarse Gravel Concrete Plastic Mix	2.976	4.168	12.40
32a	4 in. Sand and Coarse Gravel Concrete Dry Tamp Mix	3.230	4.050	13.10
33a	4 in. 100 per cent Cinder Concrete Plastic Mix	1.472	3.902	5.75
34a	4 in. 100 per cent Haydite Plastic Mix	0.943	3.960	3.73
36a	4 in. Sand and Coarse Cinder Dry Tamp Mix	2.160	3.958	8.54

TABLE 8. EFFECT OF AGGREGATE ON CONDUCTIVITY FOR MASONRY WALLS

WALL No.	DESCRIPTION OF BLOCKS USED	CONDUCTANCE C
1a	8 in. × 8 in. × 16 in. — 3-Oval Core Cinder Block	0.577
2a	8 in. × 8 in. × 16 in. — 3-Oval Core Haydite Block	0.495
3a	8 in. × 8 in. × 16 in. — 3-Oval Core Sand and Gravel Block	0.882
4a	8 in. × 8 in. × 16 in. — 3-Oval Core Limestone Block	0.856
5a	8 in. × 8 in. × 16 in. — 3-Oval Core Cinder Block	0.531
6a	8 in. × 12 in. × 16 in. — 3-Oval Core Sand and Gravel Block	0.777

TABLE 9. INSULATING VALUE OF AIR SPACES BOUNDED BY TWO PARALLEL CONCRETE SURFACES NORMAL TO THE PATH OF HEAT FLOW

WALL No.	DESCRIPTION OF WALL	CONDUCTANCE C FOR WALL	APPARENT CONDUCTANCE a FOR AIR SPACE
32a	Monolithic 4 in. Sand and Coarse Gravel. Dry Tamp Mix	3.23	...
35a	Two monolithic 4 in. Sand and Gravel. Dry Tamp Mix. Walls separated by 2.52 in. Air Space	0.660	1.11
7a	4 in. × 8 in. × 16 in. — 3-Core Cinder Partition Tile	1.003	...
8a	Two walls same as 7a spaced 1 in. apart	0.358	1.25

There are no 12-in. monolithic walls for comparison, but the 12-in. masonry walls, 5a and 6a of Table 8 show a reduction in conductance of $\frac{0.777 - 0.531}{0.777} \times 100 = 31.7$ per cent when using cinders as compared with sand and gravel, whereas for the 8-in. wall the reduction was 34.6 per cent. This rather close agreement between the 8-in. and the 12-in. walls merely shows that the core spaces have about the same relative effectiveness in both types of blocks.

The relative insulating values of the 8-in. and 12-in. masonry walls are shown by comparing the test results for 1a to 5a and 3a to 6a. Taking 1a and 5a the reduction in conductance is,

$$\frac{0.577 - 0.531}{0.577} \times 100 = 8.29 \text{ per cent.}$$

Taking 3a and 6a the reduction is,

$$\frac{0.882 - 0.777}{0.882} \times 100 = 13.5 \text{ per cent.}$$

In the case of the monolithic walls the reduction in conductivity for a 12-in. wall over that for an 8-in. wall should be 50 per cent. The reason for this poor showing for the 12-in. masonry wall as compared with the 8-in. masonry wall is due to the fact that the size and shape of the air spaces are constructed to give substantially the same value for both walls. This is apparent by comparing the dimensions and air space construction of the 8-in. and 12-in. blocks as shown in Figs. 1 and 2. The air spaces in the 12-in. block are wider along the path of heat flow than those in the 8-in. block, but they are of no greater insulating value as an air space reaches its maximum heat resistance at approximately 1 in. in width.

The results of Table 10 show the improvement in thermal conductivity obtained by placing insulation in the vertical air spaces of the test walls, and also some of the precautions which must be taken when insulating walls of this type. The percentage in improvement was based on the reduction in overall conductivity coefficients for a 15-mile wind. The apparent conductivity value of 0.33 for Rock Wool as found when comparing masonry walls 8a and 9b is slightly higher than would be anticipated, but the value of 0.478 as calculated from the test results of walls 35a and 35c is so far out of line as to require further consideration.

In all cases the test walls consisted of two vertical sections spaced 2½ in. apart and held together with tie rods which passed through the air space or insulating material. These tie rods were embedded in the concrete at both ends and provided paths of high heat conductivity. If it is assumed that the contact between the ends of the rods and the concrete was of sufficient area to supply heat to the rods at one end and remove it from the other in the same proportion as it was supplied and taken away from the air space or the insulating material in the air space, then the insulating material and the rods would each transmit a part of the total heat, which would be proportional to their heat conducting capacity. In each case this capacity would be equal to the area of the material times its thermal conductivity. If the conductivity of Rock Wool is taken as 0.27 and that of iron at 324.0, one square foot of tie rod is equal to $324.0 \div 0.27$ or 1200 sq ft of Rock Wool as a heat conductor.

TABLE 10. INSULATION PLACED IN AIR SPACES BOUNDED BY PARALLEL SURFACES

WALL No.	DESCRIPTION OF WALL	OVERALL COEFF. U 15 MILE WIND VELOCITY	PER CENT IMPROVEMENT IN OVERALL COEFF. U DUE TO INSULATION	OVERALL RESISTANCE $1/U$	GAIN IN RESISTANCE DUE TO INSULATION	APPARENT CONDUCTIVITY OF INSULATING MATERIAL AFTER CORRECTING FOR AIR SPACE
8a	Two Masonry Walls built of 4 in. × 8 in. × 16 in. 3-Core Cinder Partition Tile spaced 1 in. apart	0.279	...	3.58	...	...
8b	Same as 8a with Air Space filled with Rock Wool	0.176	36.9	5.68	2.10	0.33
35a	Two 4 in. Monolithic Sand and Gravel. Dry Tamp Mix. Concrete Walls spaced 2.52 in. apart	0.433	...	2.31	...	...
35b	Wall No. 35a. Air Space filled with Dry Cinders	0.378	12.7	2.65	0.34	2.00
35c	Wall No. 35a. Air Space filled with Rock Wool	0.150	65.3	...	4.02	0.478

In wall 8b there were 15 tie strips of 28-gage metal ¾-in. wide, giving a total area of 0.146 sq in., or the equivalent of,

$$\frac{0.146}{144} \times 1200 = 1.22 \text{ sq ft of Rock Wool.}$$

Correcting the calculated value of 0.33 gives

$$0.33 \times \frac{9 \text{ (square feet of test area)}}{9 + 1.22 \text{ (equivalent area of insulation)}} = 0.291$$

as the true thermal conductivity value for Rock Wool used in the masonry walls.

For wall 35c there were twelve ¼-in. tie rods through the test area equal to 0.588 sq in. in area, or the equivalent of

$$\frac{0.588}{144} \times 1200 = 4.9 \text{ sq ft of insulation.}$$

Correcting the calculated value of 0.478 gives

$$0.478 \times \frac{9}{9 + 4.9} = 0.31$$

as the true value of k for Rock Wool when using it in the monolithic wall.

While the assumptions as to the proportional amounts of heat conducted through the tie rods and the insulating materials are only approximately correct, the calculations based on them give reasonable values for the conductivity coefficient of the insulating material, and by this assumption calculated over-all coefficients agree reasonably well with test coefficients.

If metal tie rods, having their ends embedded in a high conductivity material, pass through an insulated area, they become important factors in the heat conductivity through that area and must be taken into consideration in making calculations. The effect of such rods becomes more important as the thermal conductivity of the space is reduced. For best results the area of the rods should be kept as low as consistent with structural requirements, and it is probable that a greater improvement in the overall coefficient for the test walls

TABLE 11. INSULATION PLACED IN CORES OF THE BLOCKS USED IN MASONRY WALLS

8 in. × 8 in. × 16 in. 3-Oval Core Cinder Blocks

WALL No.	INSULATING MATERIAL	DENSITY OF INSULATION LB/CU FT	LB INSULATION PER SQ FT WALL AREA	EQUIVALENT THICKNESS OF INSULATION OVER WALL SURFACE	OVERALL COEFF. U FOR 15 MILES WIND VELOCITY
1b	None	...	...	...	0.370
1c	Granulated Cork	5.12	1.34	3.15	0.201
1d	Dry Cinders	69.7	18.29	3.15	0.283
1e	Rock Wool	14.21	3.72	3.15	0.211

WALL No.	OVERALL RESISTANCE $1/U$	PER CENT IMPROVEMENT IN OVERALL COEFF. U FOR WALL TESTED	GAIN IN RESISTANCE DUE TO INSULATION	EFFECTIVE CONDUCTIVITY k FOR INSULATING MATERIAL
1b	2.70	...	...	...
1c	4.97	45.7	2.27	1.38
1d	3.53	23.5	0.83	3.80
1e	4.74	43.0	2.04	1.56

might have been obtained by reducing the number and size of such tie rods to minimum requirements.

The results in Table 11 show the effectiveness of insulating material when placed in the cores of concrete masonry walls. The high effective conductivities are due to the fact that large percentages of heat flow through the solid partitions of concrete between the cores which are not affected by the insulation.

The gain in heat resistance for wall 37b over wall 37a is 2.247 — 0.968 =

The effect of placing insulating material between vertical concrete studs of concrete walls is shown in the results for walls 37a, 37b and 37c, Table 12. In this table the equivalent thickness of insulation over the wall area was calculated by multiplying the total thickness of 5 in. between the studs by 20/24 to get the average thickness over the wall.

The gain in heat resistance for wall 37b over wall 37a is 2.247 — 0.968 = 1.28. A part of this gain is due to the air space and a part to the metal lath and plaster. The gain in resistance due to adding insulation in the air space

is equal to 7.812 — 2.247 = 5.56. In adding this insulation the value of the air space as such is lost. Therefore a proportional amount should be added to the gain of 5.56 in order to get the actual added resistance to be accredited to the Rock Wool as applied:

$$5.56 + \left[0.91 \times \frac{20}{24}\right] = 6.32,$$

the equivalent resistance of the Rock Wool over the full surface of the wall.

Since the equivalent thickness of insulation is 4.16 in. the effective conductivity, k, for the Rock Wool is $\frac{4.16}{6.32} = 0.658$. The explanation for the low

Table 12. The Effectiveness of Insulation When Placed Between 4 in. × 4 in. Vertical Concrete Ribs in a Wall

Wall No.	Construction	Overall Coeff. U 15 Mile Wind Velocity	Per cent Improvement in Overall Coeff. U	Equivalent Thickness of Insulation over Wall Area	Overall Resistance $1/U$
37a	2 in. Sand and Gravel Monolithic Concrete with 4 in. × 4 in. ribs. Plastic Mix	1.032	...	...	0.968
37b	Same as 37a with ½ in. Plaster on Metal Lath. Furred 1 in. on ribs	0.445	56.8	...	2.247
37c	Same as 37b with Air Space filled with 5 in. Rock Wool. Density 9.53 lb	0.128	87.5	4.16	7.812

effective insulating value for Rock Wool when used in this manner is the same as that for insulating material applied in the cores of concrete blocks. The heat which flows through the 4 in. x 4 in. ribs is not affected by the insulation. The flow is, however, restricted to a certain extent by the application of the 1 in. x 2 in. wood furring strips.

The effect of Cement Paint on the surface of masonry walls is shown in Table 13. The gain is probably largely due to the fact that it prevents the circulation of air through the porous material near the surface rather than to any insulating qualities of the paint proper.

Conclusions

1. For concrete of average aggregates without special consideration for low density, a conductivity value of 12.0 to 13.0 is reasonable.

2. Light weight aggregates are more effective in reducing the conductance of monolithic walls than they are in reducing the conductance of masonry walls with cored-out sections in the blocks.

3. The thermal conductivity of monolithic concrete is in general directly proportional to its density. Since the density is largely governed by its porosity and porosity is measured by the amount of water absorbed, the percentage of water absorption by volume is suggested as a measure of the thermal conductivity of monolithic concrete construction. This relation should have further investigation.

4. The reduction in conductance of a masonry wall built with 12-in. blocks over that of a similar wall built with 8-in. blocks depends upon the density of the aggregate, but for average construction it is less than 15 per cent, whereas

TABLE 13. EFFECT OF CEMENT PAINT ON THE SURFACE OF MASONRY WALLS

WALL No.	DESCRIPTION OF WALL	SURFACE FINISH	CONDUCTANCE C	REDUCTION IN CONDUCTANCE, PER CENT, DUE TO ADDITION OF SURFACE PAINT
1a	8 in. × 8 in. × 16 in. 3-oval Core Cinder Block	None	0.577	...
1b	Same as 1a	Two coats Waterproofed White Portland Cement Paint	0.522	9.5
2a	8 in. × 8 in. × 16 in. 3-oval Core Haydite Block	None	0.495	...
2b	Same as 2a	Two coats Waterproofed White Portland Cement Paint	0.454	8.3
3a	8 in. × 8 in. × 16 in. 3-oval Core Sand and Gravel Block	None	0.882	...
3b	Same as 3a	Two coats Waterproofed White Portland Cement Paint	0.851	3.5

for a monolithic wall the reduction would be 50 per cent. This difference is due largely to the shape and location of the air spaces in the design of the blocks.

5. In concrete masonry or monolithic walls built up with parallel slabs for the purpose of obtaining an air space between or for the use of insulating material between the two slabs, special consideration must be given to the conductivity of any tie rod or support between the slab.

6. The effectiveness of an insulating material when placed in the cores of concrete blocks will depend upon the thermal conductivity of insulation, the thermal conductivity of the aggregate, and the design of the cored spaces in the block.

7. From conclusions 4 and 5 it follows that in order to get the full value of an insulating material when placed in any wall, it must be so placed as to

effectively block off all paths of heat flow which may join the two surfaces of the wall.

8. A surface finish may serve to reduce the interchange of air through the surface and therefore reduce the thermal conductivity.

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DISCUSSION

R. E. COPELAND[1] (WRITTEN): The data presented in this paper should be of considerable interest and use to everyone concerned with the construction costs, heating economy and comfort of homes and buildings. On completion of the test program as now outlined, information on the thermal coefficients will be made available on practically all types of concrete masonry and reinforced concrete walls including the more promising new designs and lightweight concrete mixtures now being developed.

Considering the tests just reported, it appears that some of the masonry walls gave slightly better values than those obtained in previous tests of comparative walls. It may be that such differences are due to differences in the aggregates or concrete mixtures used, but apparently there is no definite evidence to that effect. Whatever the reasons, it should be noted that in making the block and building the walls for this investigation, the usual standard practices were followed throughout.

The performance of the masonry and hollow double monolithic walls with filled insulation was particularly gratifying because of the economies which the insulation method makes possible. To appreciate this more distinctly certain cost comparisons may be made. A cinder concrete block wall insulated in the usual manner with ½ in. insulating lath and plaster furred out has a thermal coefficient of around 0.20 and costs about 45 cents per square foot.

By filling the core spaces of the block with a light, porous material similar to those used in the tests, a coefficient of approximately 0.20 is obtained and the interior wall finish of either plaster or paint may be applied directly to the masonry. This wall will cost from 35 to 40 cents per square foot, or 5 to 10 cents less than the furred wall. For an average house, the savings would be at least $50 and might run as high as $175. The filled insulation has other advantages. It reduces the overall wall thickness and therefore increases the room space. It eliminates the use of combustible materials in the wall and leaves the concrete masonry surface available for use as the plaster base for which purpose it is ideal.

Similar advantages pertain to the filled insulation as applied to hollow double monolithic walls.

The test results show that insulating materials used in this manner may be relatively inefficient, because of the transmitting effects of the cross webs or metal ties. However, it appears evident from the foregoing that the effective conductivity values are not always dependable criteria of the insulation method's practical or economic worth. Of greater concern is the overall thermal coefficient and cost of the finished wall.

F. G. HECHLER[2] AND E. R. QUEER (WRITTEN): The authors are to be congratulated on the unusual completeness of the data presented in the present paper. The

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results for cinder concrete are of special interest to the writers who several years ago tested some commercial blocks made from two different kinds of cinder aggregate. Some of the results obtained supplement those presented in the paper under discussion and bring out a few relations not touched on in the paper. The tests were made with a large guarded electric hot plate between two identical walls each 4 ft x 4.5 ft with the outside faces of the walls exposed to the air in a constant temperature room. The mean temperatures of the walls varied from 55 to 95 F, therefore our results cannot be compared directly with those presented in the paper.

The first group of blocks tested was made from cinders obtained from a steel plant; when the results were higher than was anticipated a second group of blocks made from locomotive cinders was submitted. All of the values given here are for a mean wall temperature of 70 F.

The conductivity obtained from walls laid up with 4 in. solid blocks was 4.80 for the steel plant cinder blocks and 4.15 for the railroad cinders. The conductance values for 8 in. walls for blocks with two air cells were 0.51 and 0.47 respectively, and for the 12 in. blocks, 0.465 and 0.47. The poor showing of the 12 in. walls requires further comment. The industrial cinder block had four air cells arranged in pairs, with a continuous longitudinal web between. When the wall was laid no mortar was placed on this web, consequently all of the air spaces communicated freely with each other, which materially reduced the thermal resistance of the wall. The 12 in. railway cinder block had only three air spaces each extending well across the block. This gave a wall with a conductance about equal to that of the other 12 in. wall without a mortar seal on the mid rib.

It should be noted that if the air cells are not closed at the top of the wall there will be considerable convection loss. Not infrequently careless builders neglect to properly seal these walls which may result in an appreciable heat loss. Tests on the 8 in. wall gave a conductance 33 per cent greater for an open wall than for the same wall with the air spaces closed off; for the 12 in. wall the value was 32 per cent greater. Although the authors do not state whether the air spaces in their walls were sealed, we assume that they were, which would be inferred from the test method used.

The authors mention the difficulty of measuring true surface temperatures on the walls because of the variation in resistance of hollow and solid material. Surface irregularities on materials like cinder block also make it difficult to attach thermocouples so that they read true surface temperatures. Covering the outer surface with paper or a skim coat of cement invariably gives a slightly higher conductance value than the plain wall alone.

We have also from time to time tested some 3 ft x 3 ft concrete slabs 4 in. thick that are reported to have been made about 1920 but for which no records are available as to composition. They appear to be an ordinary good grade of sand and crushed limestone aggregate and weigh about 130 lb per cubic foot. Test results over a period of years gave conductivities, at a mean temperature of 80 F, varying from 12 to 13, and we believe that for design purposes a value of $k = 12$ is desirable. However, we have sometimes obtained much lower values; for example, some test specimens 12 in. x 12 in. x 1.5 in. made of a 1-2-3 mix and weighing 144 lb per cubic foot gave a value of 8.7.

W. A. DANIELSON: I happen to get the building end of the game as well as the heating. The last speaker said to fill up the joints. It is the last thing that should be done if a dry wall is desired. Cinder blocks laid up with no through mortar joints will result in a dry wall. Capilarity is more important than porosity. Cinder blocks are more porous, but when set in a pan of water the water is not drawn up by capilarity. If building materials are such that moisture passes through, either as free water or by capillary action, trouble results.

A house built out of concrete recently in Washington, and with a very porous insulator on the inside, has so much condensation running down the inside during cold weather that it forms pools on the floor. The breathing of the wall carries warm air through the insulation into contact with the cold concrete and condensation troubles start. Don't forget in discussing heating that the place must be lived in or used, and moisture is one of the most important factors in the result. I do not believe that moisture received the consideration it should have received in the paper and the discussion that followed.

F. B. ROWLEY: Colonel Danielson has referred to the effect of moisture on the thermal conductivity. This is, of course, an important factor and while it has been omitted in the work recorded in this paper it is taken into consideration in that part of the program which follows. Certain walls are being tested dry and then soaked with water for various lengths of time, and retested to determine whether or not the moisture will effect the conductivity.

I would like to say that during this test program we have had the fullest cooperation of the Research Laboratory and of the *Portland Cement Association*, and it is only through such cooperation that it is possible to carry an investigation of this type.

MR. COPELAND: We have just completed some tests of the rain resistance of various types of concrete masonry walls. These walls were subjected to an artificially simulated rainstorm consisting of a 25 mph wind and 2½ in. rain intensity. This investigation showed that even the porous types of concrete masonry will successfully stand up under 24 hours of this severe test, if the exterior surface is thoroughly painted with a waterproof Portland cement paint or is stuccoed. The majority of concrete masonry walls in buildings are either painted or stuccoed, and if such treatments are properly put on, they prevent any trouble from moisture coming through the concrete block.

No. 1027

COMPARATIVE STUDY OF COMBUSTION RESULTS WITH VARIOUS THERMOSTATS

By Burton E. Shaw * (*MEMBER*), Des Moines, Iowa

THERE has recently been a decided trend in the heat control industry toward a widespread application of thermostats built by different manufacturers employing artificial heating within the thermostat itself. Various ingenious methods of artificial heat application have been developed, but it is not the purpose of this paper to discuss the possible advantages or disadvantages of these devices. In the results obtained all thermostats within the same general classification or category performed to produce practically the same results. In this paper eight different thermostats of three manufacturers were investigated. Operating results were obtained on all of these thermostats, but because of limited time and test facilities, as well as the closely allied performance data which were found, certain thermostats were investigated more fully than others.

For the purpose of this paper, thermostat types *A, B, C, D,* and *E* were of artificially heated construction; type *F* was a standard precision-construction conventional unit; type *G* was of the same construction except that it employed a clock used for set-back temperatures at night. Type *H* was a clock-actuated thermostat which operated at fixed time intervals to turn on the heat source (in this case an oil burner), if any room temperature drop of a fractional degree occurred at the thermostat location. Results obtained on this latter instrument as far as the temperature control was concerned, indicated that it operated over the period tested identically to that of any of the artificially heated types of thermostats arranged for so-called short cycling operation.

It is the purpose of this paper to correlate the results obtained on the various thermostats mentioned in order to compare the effect of thermostat design, operation, and location on various factors affecting flue gas loss, oil consumption, off-period stack loss, and the most important factor of all—physiological comfort. It is the further purpose of this paper to report upon the effect on oil consumption of lowering the thermostat setting during the night. It must be borne in mind that the results reported herein are those encompassed only by this investigation, and may not generally apply to other installations.

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Presented at the 42nd Annual Meeting of the American Society of Heating and Ventilating Engineers, Chicago, Ill., January, 1936.

RESIDENCE A

During the heating season of 1934-35 a large well constructed eight-room frame house, located in Southwick, Mass., having a gravity warm air system installed, was thoroughly investigated. For a design inside temperature of 70 F, and 0 F outside, the calculated hourly Btu net load was 106,000 Btu. The standard Btu loss method of calculating this value was later checked by computed results at 65 degree-days. In this residence calibrated thermometers were located at the floor, 30 in. and 5 ft above the floor and at the ceiling level in the living room adjacent to the thermostat. Thermometers were also occasionally read at the top of the second floor staircase in order to determine possible temperature over-run. A two-bulb standard recorder was placed adjacent to the thermostat to record room temperatures. The bulbs at different times were arranged to record floor temperatures, 30 in. zone temperatures, 5 ft and ceiling temperatures. After an initial study relative to temperature variations the 24-hour chart instrument was replaced by a 7-day instrument set to record temperatures at the thermostat location. An outside temperature recorder covering a range of —30 to + 120 F was installed to record outside temperatures. No local weather bureau data were available and it was obvious that all results in order to be of value should be based on degree-days. A stack temperature recorder was located at the breeching of the furnace. A blower system was installed in conjunction with the gravity warm air furnace in order to convert the heating plant from a gravity to a forced air system. The nozzle installed in the oil burner was calibrated several times during the course of the investigation. In conjunction with the calibrated nozzle the outside 1000-gal tank was checked daily for oil consumption.

A preliminary investigation at Southwick disclosed the fact that the heating system was not in balance. In order to get a fair comparison of the various thermostats it was deemed advisable for the purposes of this paper to operate the blower at a reduced speed and to balance the system by throttling dampers located in each basement duct. This was accomplished so that a fair basis of comparison for all of the thermostats tested could be determined.

Early in the progress of the investigation it was evident that stack gas temperature alone could not be considered as a criterion of possible flue loss, even though the burner always operated at the same continuous efficiency and flue gas loss. The continuous fired carbon-dioxide concentration was set at 10 per cent with a draft of 0.02 in. water in the fire-box. In this case the setting of 10 per cent continuous carbon-dioxide concentration was selected as it is gradually becoming considered as a standard by the oil-burner industry. Spot samples were taken at different intervals for the various firing conditions, but these likewise were not completely satisfactory because of personal error and other variations. It was finally decided to continually aspirate flue gases from the time the burner started until the instant it shut off, for only in this way was it possible to obtain a satisfactory cross-section of combustion results. Gas samples initially taken were analyzed for carbon-dioxide, oxygen, and carbon-monoxide by a standard Orsat apparatus. However, it was evident that losses other than those that could be determined by a standard Orsat apparatus might be present, and it was therefore decided to secure a Bureau of Mines Burrell in order that an analysis could be made for the possible presence of hydrogen, as well as unconsumed hydro-carbons such as ethane and methane.

Also by using the high temperature combustion method, a more accurate determination of the possible carbon-monoxide constituent could be determined. A continuous aspirating device was designed similar to the set-up as reported upon by the University of Illinois. It will be noted that electrically operated solenoid valves were arranged so that they connected to the burner stack control and permitted the introduction of a gas sample only during the burner operation.

Knowing the average stack temperature conditions as well as the average constituents of the flue gas during the *on* period of the burner, it was possible to determine the average flue loss for various methods of operation. In Southwick, thermostat types *A, B, C, E, F,* and *G* were investigated. In this residence a pressure atomizing burner was employed using a calibrated nozzle delivering 3.7 gal per hour. The calculated continuous burner efficiency was 65 per cent.

TABLE 1. FUEL CONSUMPTION COMPARISONS WITH VARIOUS THERMOSTATS AND THERMOSTAT SETTINGS

DEG-DAYS	TYPE	SETTING [a]	GAL PER DEG-DAY	NO. DAYS
		RESIDENCE A		
33.9	*B*	*sc — N N S B*	0.53	10
50.3	*C*	*sc — N N S B*	0.54	21
25.6	*A*	*sc — N N S B*	0.56	3
36.5	*F*	*lc — N N S B*	0.51	19
24.1	*C*	*lc — N N S B*	0.52	2
		RESIDENCE B		
18.6	*E*	*sc — N N S B*	0.54	8
26.6	*H*	*sc — N S B*	0.50	2

[a] *sc* — Short cycling operation.
lc — Long cycling operation.
N N S B — No night set-back.
N S B — Night set-back.

RESIDENCE B

Residence *B* located in Adel, Iowa, was of practically the same construction as the one in Southwick, Mass. The investigation was continued with thermostats *D* and *H* used on the heating system which was in this case a hot water installation. Another pressure atomizing oil burner of practically the same description, but built by a different manufacturer than the one used at Southwick, was installed. This burner had a calibrated nozzle capacity of 3.16 gal per hour at a calculated efficiency of 55 per cent. The calculation of the hourly Btu requirements at Residence *B* based on 70 F inside temperature, and an outside temperature of 0 F gave the net requirement of 115,000 Btu per hour. This value was corrected to check with the actual requirements of Residence *B* as determined in the course of the investigation. In continuing the investiga-

tion in Residence *B* the main point of interest was the possible variation in results obtained by a hot water system in comparison to a warm air system. Test determinations were made in this residence in the same manner as those of Residence *A* except that in this case the burner was not set for 10 per cent carbon-dioxide concentration for continuous firing nor was an attempt made to maintain the draft constant over the fire. The installation may be considered an average one for this portion of the country, with the possible exception that no draft regulator was installed as in Residence *A*. However, it has been found that many installations like this are made each year, and it was therefore considered to be of interest to test the installation as made by an average dealer for an average application. As time permits, the investigation at Residence *B*

TABLE 2. FUEL CONSUMPTION COMPARISONS WITH VARIOUS THERMOSTAT SETTINGS

Average Deg-Days	Setting	Gal per Deg-Day per 100,000 Btu per Hour Net Requirements[a]	Gal per Deg-Day	No. Days
Residence A: 106,000 Btu per Hour Radiation, Continuous Burner Efficiency 65 Per Cent				
27.3	*sc — N S B*	0.434	0.46	3
28.7	*sc — N S B*	0.425	0.45	3
43.2	*sc — N N S B*	0.510	0.54	34
35.7	*sc/lc — N N S B*	0.500	0.53	9
55.3	*lc — N N S B*	0.480	0.51	21
Residence B: 115,000 Btu per Hour Radiation, Continuous Burner Efficiency 55 Per Cent				
26.6	*sc — N S B*	0.435	0.50	2
18.6	*sc — N N S B*	0.470	0.54	8

[a] These average rates of consumption should not be compared to rates obtained from Fig. 1 due to the greater range of outside temperature encompassed by this table.

will be continued with the oil burner set for a 10 per cent carbon-dioxide concentration, and 0.02 in. of water draft at the fire-box. At the present time on continuous firing the carbon-dioxide concentration is 5.8 per cent with the draft over the fire varying from 0.08 to 0.13 in. of water depending upon the outside temperature and wind conditions. Another reason it was considered advisable to carry on the investigations at Residence *B* under the above conditions, which might be deemed rather inefficient otherwise, was that by virtue of the results obtained at Residence *A* it was considered advisable to thoroughly study the question of *off* period sensible heat loss. This factor will be enlarged upon further in this paper.

MOUNTING, LOCATION AND CYCLING OF THERMOSTAT

At Residence *A*, comparative studies were made of the location of thermostats at the 30 in. zone and the 5 ft zone. At Residence *B* the thermostats were only compared at the 30 in. zone.

The various thermostats tested had different means of adjusting the mechanical differential of the thermostat in order to control the burner cycle, that is, the length of the burner *on* period for any one temperature condition outside. Rather than to enumerate and compare the specific settings of the different differential adjusters of the various thermostats, it seems advisable to segregate the operation by results alone. Therefore, for the purpose of this paper, the thermostats having a setting such that the room temperature variation at the thermostat was less than 1 F were termed short cycling thermostats and were referred to by the designation, *sc*. Short cycling resulted in *line temperature control* with normal variations less than ¼ F at the thermostat location. A thermostat set so that it resulted in a 1 to 2 F maximum variation at the thermostat location was called a medium cycling thermostat and was designated by the symbols, *sc/lc*. Medium cycling resulted in normal variation not in excess

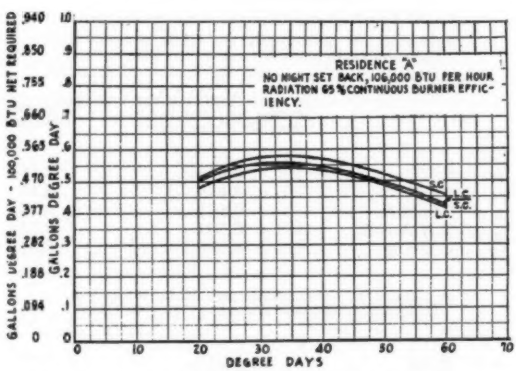

FIG. 1. FUEL CONSUMPTION CURVES FOR VARIOUS THERMOSTAT SETTINGS

of 1 F and may be termed short cycling, except for the purpose of this paper it was necessary to make a distinction between the *close temperature control* and *line temperature control* obtainable by short cycling (*sc*). Thermostats operating to maintain a maximum variation at their location of greater than 2 F are designated as long cycling thermostats and will be referred to by the symbol, *lc*. The conventional temperature recorder is scarcely more accurate than ½ F, and further, many abnormal conditions such as prolonged opening of outside doors, solar radiation on the thermostat, etc., are bound to occur occasionally to upset the plans of the investigator. Hence in reporting the results of these tests the average conditions have to be considered. The term *short cycling* as previously defined may therefore be considered a normal temperature control with approximately ¼ F or less, and the term *medium cycling*—control within approximately 1 F.

RESULTS

Thermostat Type

Based on temperature control at the thermostat location the various artificially heated thermostats performed practically alike when adjusted to give the three

types of operation previously enumerated as short cycling, medium, and long cycling. Whether or not there was any relative merit to various types of thermostat construction as regards ruggedness, or snap of contacts, is beyond the scope of this paper.

Thermostat Location

Relative to oil consumption and fuel economy all results obtained indicated that there was little or no difference to be gained by the location of the thermostat (Table 3). From the standpoint of physiological comfort, however, there seemed to be a marked difference if the reactions of the author, his family, and his friends may be considered as a safe criterion. Thermometer readings showed that floor temperatures increased on the average from 1 to 5 F depending on outside weather conditions, when a 30 in. thermostat setting was used as compared to a 5 ft location. Ceiling temperatures in line with the thermostat location were increased the same amount; however, temperatures taken at the head of the second floor staircase showed very little difference. Based on the opinion of a number of people who expressed judgment on the results obtained at Residence *A* it is the author's conclusion that the location of a thermostat at approximately the 30 in. zone will give superior comfort conditions as compared to one placed in the conventional 5 ft or breathing line location. Considering the results obtained as well as experimentation with a base board mounting it is questionable whether such a low location would be satisfactory except where the residence is completely air conditioned and well insulated. A study of Table 3 shows no difference in the recorded average gallons per degree-day for Residence *A* for short cycling operation with no night set-back (*sc-N N S B*). Although the results were averaged for a different number of days and the degree-days average also was different, it is felt that the two values (gallons per degree-day) are closely comparable. Ref-

TABLE 3. FUEL CONSUMPTION COMPARISONS WITH DIFFERENT THERMOSTAT HEIGHTS

Height	Setting	Gal per Deg-Day	No. Days	Average Deg-Days
5 Ft	*sc — N N S B*	0.54	13	35.8
30 In.	*sc — N N S B*	0.54	21	45.5
5 Ft	*lc — N N S B*	0.51	6	34.5
30 In.	*lc — N N S B*	0.52	15	35.5

TABLE 4. FLUE LOSSES WITH DIFFERENT THERMOSTAT SETTINGS

Setting	Per Cent Loss	No. Days	Ave Deg-Day
sc	25.3	4	23
lc/sc	27.6	4	21.3
lc	30.5	7	21.9

erence to Table 3 and to Fig. 1 shows that based on total averages the gallons per degree-day recorded, decreased slightly from 35.8 degree-days (*sc-N N S B-5* ft) to 45.5 degree-days (*sc-N N S B-30* in.) indicating that oil consumption for this type of operation may show a saving for the 30 in. thermostat location. It also appears from these results that the shorter the thermostat cycling the more favorable a 30 in. mounting becomes. However, there can be no doubt that on the installations investigated the *Comfort Zone* thermostat mounting resulted in no greater oil consumption and much better comfort conditions were obtained.

Length of Cycle

The effect on oil consumption of long, medium and short cycling is shown in Fig. 1, where gallons of oil consumed per day per 100,000 Btu net requirements in Residence *A* (0 vs 70 F at a calculated efficiency of 65 per cent) were plotted as ordinates against degree-days as abscissae. Also gallons of oil per degree-day were plotted as ordinates. It is evident from Fig. 1 that medium cycling as already defined in this paper will result in practically no greater oil consumption than long cycling. Straight line temperature control while the ultimate in human comfort will be somewhat less economical than long cycle control. In other words at the present stage of the art, and considering the type of oil burners investigated, *line temperature control* with about ¼ F maximum variation will be more expensive depending upon outside weather conditions, than medium or long cycle operation. The average oil consumption will be less than 2 per cent greater for medium cycling than for long cycling. The value for straight line temperature control will be slightly more than this value. Based on Fig. 1 there seems to be little difference between medium cycling control that may readily be obtained with the new type of artificial heating thermostats as compared with the long cycling method of operation obtained with former conventional thermostats. In fact at Residence *A* as outside temperance conditions became more severe with an increasing number of degree-days, the medium cycle operation holding the average temperature within a variation of less than 1 F closely approached the economy of the conventional long cycle thermostat operation. Investigation was made of the cause of extra loss on very short cycling operation with the results as shown in Tables 4 and 5. It is evident from Table 4 as well as Fig. 8 that the total flue loss increases directly with the length of cycling in the range tested. Hence, considering all radiation loss as practically a constant, greater economy with short cycling would be expected if it were not for the higher *off* period loss as shown in Table 5. *Base loss* contributing as it does to both radiation and convection losses was kept at a minimum with insulation behind the bricking and the use of light weight porous insulating brick. Convection effects were undoubtedly reflected in a somewhat higher sensible heat dry stack loss during the *on* period as well as a higher sensible heat loss during the *off* period. The *off* period loss is controllable and more cognizance should be given to this factor. The *off* period losses were calculated as shown in Appendix *A* and will be more fully discussed later. The values calculated by this method check closely with the total over all loss indicated by the oil consumption data, and the net radiation requirements of the house. They are, therefore, believed to be close to the actual conditions existing and serve their purpose of indicating the variation

of *off* period loss with length of cycle and outside temperature conditions. Table 5 also analyzes the unconsumed flue gas loss indicating that it is greater for shorter cycling, but the per cent variation is small. Furthermore the lower average stack gas temperature counteracts this slight increase to maintain the total loss less for short cycling. Fig. 2 shows degree-days as ordinates plotted against gallons per day as abscissae for the various types of thermostat operation. The curves indicate little variation between long cycling and medium cycling operations.

Table 1 shows the various types of thermostats with different settings, and the gallons per degree-day recorded at Residence *A*, for the number of days in which the investigations were made. In Fig. 1 all of the thermostats operated at their day settings continually 24 hours a day; and no night set-back was used, hence the designation *N N S B*. It is to be noted that for similar settings the results obtained on each of the thermostats were quite consistent considering variation in degree-days and the number of days in which the values were averaged. Table 1 also shows a small saving in oil consumption for long cycling vs short cycling operation.

Table 5. Stack Gas Loss Averages

Thermostat Setting	*sc*	*sc/lc*	*lc*
Average Deg-Days	25.2	20.2	26.3
Number of Days	4	4	7
Stack Temperature			
Burner Off	243 F	183 F	189 F
Burner On	309 F	385 F	445 F
Carbon-Dioxide	4.7%	5.2%	5.3%
Oxygen	11.8	11.5	11.9
Flue Gas Losses			
Moisture	7.61	7.91	8.02
Unconsumed			
Carbon-Monoxide	0.70	0.19	0.09
Methane	0.78	0.23	0.016
Ethane	1.18	0.12	0.06
Hydrogen	0.25	0.27	0.32
Total Unconsumed	2.91	0.71	0.486
Dry Flue Gas Loss	14.8	18.2	21.5
Total Flue Gas Loss	25.3	27.2	30.4
Off-Period Loss	35.0	27.6	25.6
Total Loss	60.3	54.8	55.3
Gal Fuel per Deg-Day	0.55	0.50	0.52

Table 2 compares various types of thermostat operation on the different thermostats, with gallons per degree-day, gallons per degree-day per 100,000 Btu net heating requirements at Residence *A*, and the number of days during which each instrument was tested. Table 2 also shows a group of values obtained at Residence *B* on a hot water installation over a period of eight days in which thermostat *D* was set for short cycle operation with no night set-back. It will be noted that the gallons per degree-day were lower as recorded at Residence *B* than the gallons per degree-day at Residence *A* (*sc-N N S B*), however the degree-days were also lower. The gallons per degree-day per 100,000 Btu net requirements at Residence *B* were less than Residence *A*

with a thermostat setting for no night set-back showing some increased efficiency in the hot water plant over the warm air installation even though the actual flue gas efficiency of the burner at Residence *B* was less than it was at Residence *A*. This may also be explained by a variation in the number of degree-days in each case (Fig. 9). These data are not to be construed as indicating that the hot water plant is intrinsically more efficient than the warm air system; for there are several other factors involved. Actually the hot water installation was more efficient than the old converted warm air system used at Residence *A*.

NIGHT SET-BACK

Relative to the effect of night set-back on oil consumption Table 2 shows a marked saving in oil consumption based on gallons per degree-day for night set-back designated *N S B*. This saving was 7 per cent for short cycle opera-

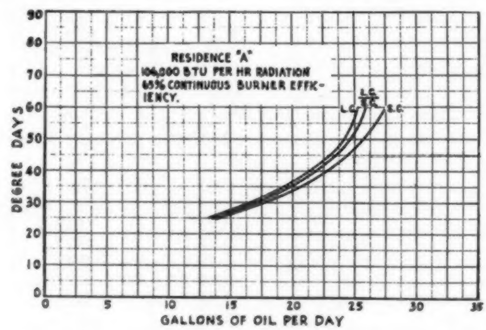

FIG. 2. VARIOUS THERMOSTAT FUEL CONSUMPTION CURVES PLOTTED AGAINST DEGREE-DAYS

tion at Residence *B* for approximately 22 degree-days, and nearly 12 per cent for long cycle operation at Residence *A* compared, however, for 28.7 degree-days against 35.3 degree-days. Additional saving for long cycle operation can be explained by lower *off* period losses. Table 9 shows the effect of night set-back on average stack gas temperatures for various types of operation.

FLUE LOSS

The calculated *off* period sensible heat loss has been shown in Tables 6, 7, and 8 for each of the types of operation and it is to be noted that they are highest for short cycle operation, and lowest for long cycle operation. If these latter calculated losses are considered in conjunction with the flue-gas loss during the *on* period, the total loss assumes a high value. The calculated losses, however, compare with the results actually obtained and it would, therefore, seem that *off* losses assume a larger value than various investigators have previously reported. These losses are controllable by application of air inlet dampers to be shut during the *off* period of the burner, or by use of a boiler

TABLE 6. OFF-PERIOD LOSSES NO NIGHT SET-BACK, 20 DEGREE-DAYS

THERMOSTAT SETTING	*sc*	*lc/sc*	*lc*
Average Temperature T (F)	273	185	185
Velocity of Air (ft per sec)	8.2	7.8	7.8
Volume (cu ft per sec)	2.87	2.73	2.73
Time Off (sec)	73,600	74,600	73,000
Volume (cu ft per day)	210,000	204,000	199,000
Specific Gravity	0.00086	0.00100	0.00100
Weight (lb per day)	11,250	12,700	12,500
Temperature Difference (T-50 F)	223	135	135
Btu per Day	626,000	430,000	423,000
Average Degree-Days	23.0	21.3	21.9
Fuel (lb per day)	92.2	80.5	84.6
Btu Lost per lb Fuel	6,790	5,350	5,000
Per Cent Off-Period Loss	35.0	27.6	25.6
Btu Lost per Degree-Day	27,200	20,200	19,200

TABLE 7. OFF-PERIOD LOSSES NO NIGHT SET-BACK, 40 DEGREE-DAYS

THERMOSTAT SETTING	*sc*	*lc/sc*	*lc*
Average Temperature T (F)	390	251	192
Velocity of Air (ft per sec)	9.7	8.1	7.73
Volume (cu ft per sec)	3.40	2.84	2.72
Time Off (sec)	41,200	51,000	59,160
Volume (cu ft per day)	140,000	145,000	161,000
Specific Gravity	0.00075	0.00090	0.00095
Weight (lb per day)	6,500	8,130	9,520
Temperature Difference (T-50 F)	340	201	142
Btu per Day	555,000	410,000	338,000
Average Degree-Days	39.8	40.7	40.7
Fuel (lb per day)	181	163.0	151.0
Btu Lost per lb Fuel	3,060	2,510	2,240
Per cent Off-Period Loss	15.8	13.0	11.6
Btu Lost per Degree-Day	13,900	10,100	8,300

TABLE 8. OFF-PERIOD LOSSES NO NIGHT SET-BACK, 60 DEGREE-DAYS

THERMOSTAT SETTING	*sc*	*lc/sc*	*lc*[a]
Average Temperature T (F)	465	299	235
Velocity of Air (ft per sec)	11.0	9.2	8.4
Volume (cu ft per sec)	3.85	3.22	2.94
Time Off (sec)	32,200	43,800	46,200
Volume (cu ft per day)	124,000	141,000	136,000
Specific Gravity	0.00069	0.00084	0.00090
Weight (lb per day)	5,340	7,400	7,610
Temperature Difference (T-50 F)	415	249	185
Btu per day	555,000	461,000	352,000
Average Degree-Days	60.2	62.7	50.3
Fuel (lb per day)	193	195	169
Btu Lost per lb Fuel	2,890	2,360	2,100
Per Cent Off-Period Loss	14.9	12.2	10.8
Btu Lost per Degree-Day	9,200	7,530	7,000

[a] *lc* — 50 Degree-Days.

flue-gas discharge damper controlled by the operation of the burner. It is apparent from this investigation that much may yet be accomplished toward increased economy in the operation of oil burning appliances.

Tables 6, 7, and 8 show calculated *off* period losses considering no night set-back for 20, 40 and 60 degree-days respectively. These calculations were based on an average gas temperature as shown in the first column of the table, on an average outside temperature based on the various degree-days, with a total chimney height of 30 ft, considering an 8 in. flue pipe and with a draft based on the calculated average draft over the fire box as shown in Fig. 3, where draft in chimney and stack was plotted vs degree-days for long, medium and short cycling. Fig. 4 shows the calculated stack velocity in feet per second vs degree-days for different types of thermostat operation.

It is evident that the *off* period loss increases inversely as the length of the burner cycle, and that the *off* period loss for any one type of thermostat operation decreases with an increase in degree-days. This latter condition is further shown in Fig. 6 showing thousands of Btu per degree-day *off* period loss plotted vs degree-days.

TABLE 9. EFFECT OF NIGHT SET-BACK ON STACK TEMPERATURES

	N S B TEMP F	AVERAGE DEGREE-DAYS	*N N S B* TEMP F	AVERAGE DEGREE-DAYS
Burner *off*:				
sc	213	26.5	273	23.8
lc/sc	178	17.0	185	21.3
lc	166	27.0	185	24.0
Burner *on*:				
sc	346	26.5	273	23.8
lc/sc	393	17.0	283	21.3
lc	415	27.0	456	24.0

Table 9 shows the effect of night set-back on average temperatures during the burner *off* period as well as the burner *on* period. Table 9 is compared for similar weather conditions of approximately 20 degree-days. Weather conditions at both Residences *A* and *B* throughout the heating season averaged between 20 and 30 degree-days per day.

In an attempt to more closely study the *off* period sensible heat losses Fig. 3 was drawn indicating actual conditions obtainable at Residence *A* showing variations in chimney draft in inches of water plotted against degree-days for long and short cycling.

Fig. 4 indicates the average velocity of air traveling through the stack during the *off* period at Residence *A* based on long, medium, and short cycle operation, where velocity is plotted against degree-days. Fig. 4 is based on the draft over the combustion chamber as determined from Fig. 3 as well as average temperature conditions obtained for short and long cycling for the various degree-days.

Actual conditions obtained at Residence *B* are shown in Fig. 5 with no draft adjuster installed. It is to be noted that the combustion chamber draft is higher than for Residence *A*, which was adjusted for 0.02 draft, and that

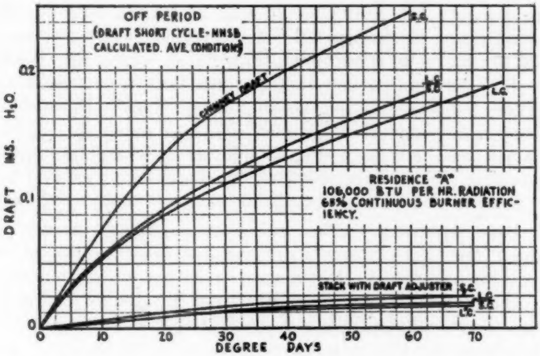

FIG. 3. CHIMNEY DRAFT CURVES FOR VARIOUS THERMOSTAT SETTINGS, *Off* PERIOD OPERATION

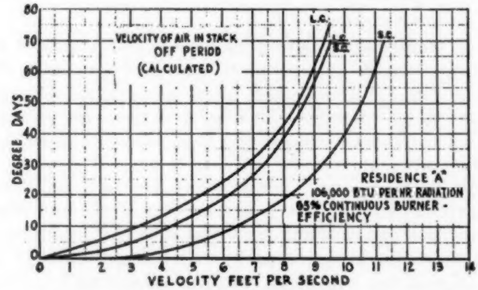

FIG. 4. VELOCITY OF AIR IN STACK FOR *Off* PERIOD OPERATION WITH VARIOUS THERMOSTAT SETTINGS

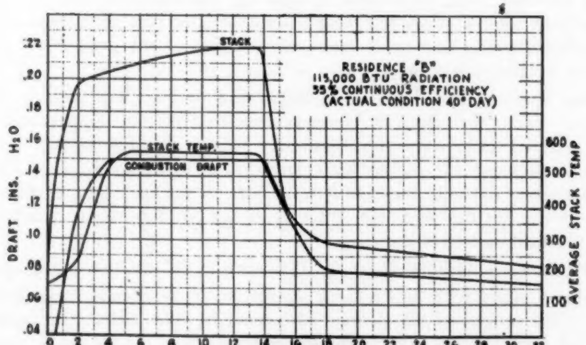

FIG. 5. STACK TEMPERATURES AND COMBUSTION DRAFT FOR A PERIOD OF OPERATION ON A 40 DEGREE-DAY

the *off* period losses for Residence *B* will be higher than for Residence *A*. The *off* period losses as determined for Residence *A* are shown in Tables 6, 7, and 8, and are based on conditions prevailing at Residence *A* with the standard setting of 10 per cent carbon-dioxide continuous fired concentration. However, Residence *A* *off* period losses as shown in Tables 6, 7, and 8 are corrected for the variation in combustion chamber draft as shown in Fig. 3. Based on considerable test data available, Fig. 3 indicates a close approximation of the draft changes that may be expected over the combustion chamber for variations in external weather conditions when a conventional stack draft damper is employed.

Actual draft and stack temperature conditions prevailing at Residence *B* are shown in Fig. 5 on a 40 degree-day (average outside temperature 25 F) with these various factors plotted against time in minutes during one complete burner cycle. From these curves it is evident that a high combustion chamber draft was prevalent for this installation when no draft regulator was employed. Although these conditions may not be considered as an average, there are many installations where no draft controls are used, and it is possible that in these installations the *off* period loss is greater than that shown in Tables 6, 7, and 8 calculated for Residence *A*.

Average stack temperatures during *off* period and length of time *off* plotted for long and short cycles of operation against degree-days at Residence *A* are illustrated in Fig. 7. The average stack temperature during *off* period for long cycle firing based on degree-days is lower than the average temperature during *off* period for short cycle firing. It is apparent from these curves that the time *off* for the same number of degree-days is greater for long cycle firing than with short cycling. Furthermore the time *off* decreases with an increase in degree-days to a point where the long cycle and short cycle curves converge. Zero time *off*, or continuous operation will occur at the number of degree-days at which the burner capacity is set to deliver 70 F inside temperature for the degree-day equivalent outside temperature. The temperatures plotted in Fig. 7 are the temperatures in the stack during the *off* period, thus indicating a reason for the greater *off* period loss for short cycle operation than for long cycling. The difference in time *off* for the same number of degree-days for both long and short cycling is due to the fact that there is a higher overall efficiency for long cycle operation because of the greater *off* period loss with short cycling operation.

A 7-day chart at Residence *A* is illustrated in Fig. 9 indicating long cycle operation. From this chart it is interesting to note the temperature over-run which occurred at 2 p.m. on Thursday when the thermostat setting was changed to increase the temperature from 68 F to an average of 73 to 74 F. The temperature actually increased to 76 F resulting in a temperature over-run of 2 F and a further over-run may be noted at 3 p.m. on Sunday when a similar condition occurred.

Fig. 10 shows a 24-hour chart at Residence *A* of medium cycle operation.

A 24-hour chart with short cycle operation at Residence *B* is illustrated in Fig. 11. The length of the burner *on* period is indicated at the outer circumference of the chart by a drop in the exterior line drawn by a pen actuated from a relay wired to the burner motor. It may be noted from this chart that practically straight line temperature conditions were obtainable. The

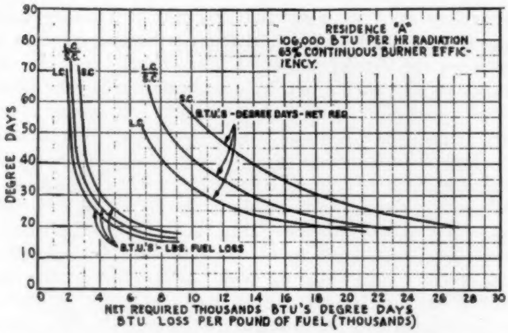

FIG. 6. NET BTU REQUIREMENTS AND FUEL LOSSES PLOTTED AGAINST DEGREE-DAYS

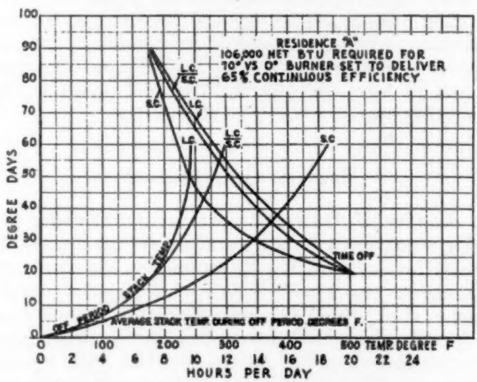

FIG. 7. *Off* PERIOD STACK TEMPERATURES AND PERIODS OF BURNER OPERATION PLOTTED AGAINST DEGREE-DAYS

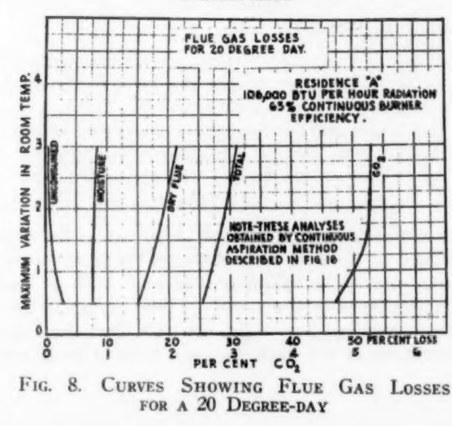

FIG. 8. CURVES SHOWING FLUE GAS LOSSES FOR A 20 DEGREE-DAY

deviations occurring at 8:30 a.m. and at 4:20 p.m. were caused by prolonged opening of an outside door. As regards human comfort there is little comparison between the results shown in Figs. 9, 10, and 11. It is apparent from the results obtained that short cycle operation gives the ultimate in comfort in the room where the thermostat is located. Of course it is evident that extremely close temperature control at the location of the thermostat without proper balance and distribution of the heating system may result in unsatisfactory heating at remote points. If a system is out of balance it is frequently desirable to forego the comfort obtainable in the living room or at the thermostat location in order that better distribution may be accomplished, and in such cases longer cycling would have to be resorted to. A thermostat should not be expected to rectify major heating plant faults, but it may assist in their correction, and it should be recognized that balanced plants are desirable in order that the ultimate in comfort may be obtained with artificially heated thermostats.

Figs. 12, 13, and 14 give a comparison of stack temperatures for long, short, and medium cycling for 20 degree-day operation. Fig. 12 shows the stack temperature condition existing for long cycle operation, and an average outside temperature of 45 F (20 degree-days). The stack temperature conditions for short cycle and 20 degree-day operation are illustrated in Fig. 13. Fig. 14 shows medium cycling operation for a 20 degree-day.

MORNING OVERSHOOTING

The effect of the type of thermostat operation on morning overshooting for Residence *A* using a converted forced warm air system is illustrated in Fig. 15. On a 40 degree-day for short cycling operation Fig. 15*a* shows no overshooting occurred after a night set-back of 6 F on a 17 degree-day, and Fig. 15*b* shows 1-F overshooting. High speed blower operation (increased air circulation, and hence less *stratification*) tended to reduce the room temperature variation from 1 to ½ F as indicated in Fig. 15*b*. Good long cycle operation with only 2 F overshooting on a 25 degree-day, is shown in Fig. 15*c*, however, overshooting as high as 4 F was experienced with this type of operation.

The gas sampling apparatus employed for the determination of flue gas samples is described in Fig. 16.

Table 9 shows the effect of night set-back on stack temperatures for a 20 degree-day. The stack temperature during the *on* period averages lowest with the shortest cycle, both for night set-back and no night set-back. However, the average stack gas temperature is lower with no night set-back than it is with night set-back with the same outside temperature conditions. Stack temperatures during the *off* period of the burner are higher for the shortest cycling and are also higher for no night set-back than they are for night set-back.

The higher stack temperatures during the *on* period where night set-back was employed is occasioned by the fact that in the morning when the room thermostat calls for burner operation to bring the room back to normal, the stack temperature rises above its normal conditions, thus giving in the 24-hour period a higher temperature average for night set-back than for no night set-back. This average temperature in the stack does not prevent more economical opera-

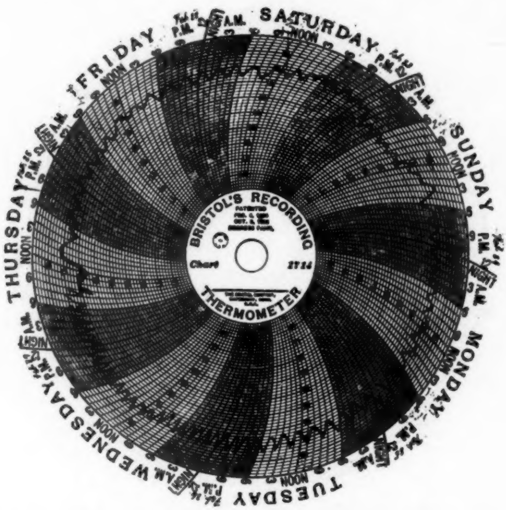

FIG. 9. RESIDENCE *A* INSIDE TEMPERATURES, *LC* OPERATION

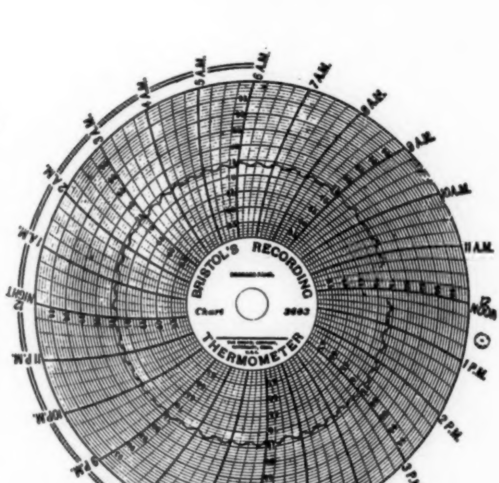

FIG. 10. RESIDENCE *A* INSIDE TEMPERATURES, *LC/SC* OPERATION

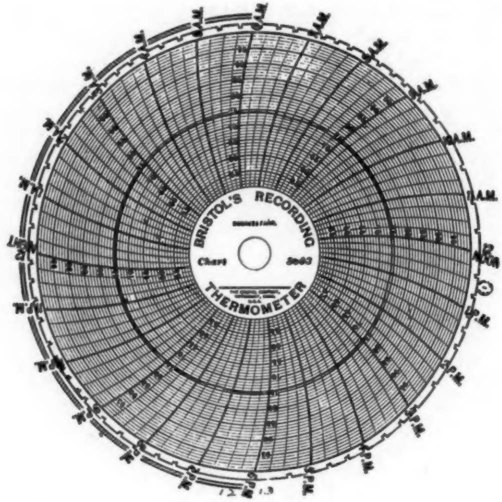

FIG. 11. RESIDENCE *B* INSIDE TEMPERATURES, *SC* OPERATION

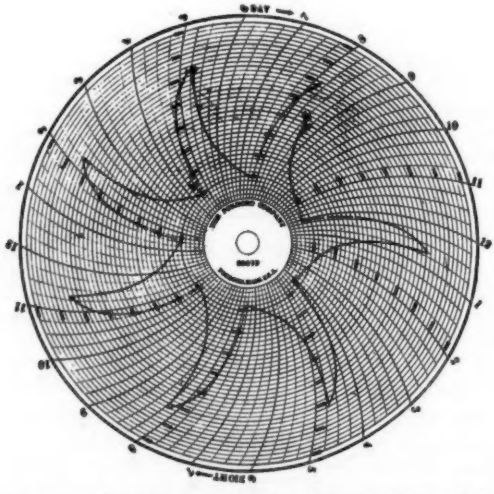

FIG. 12. RESIDENCE *A* STACK TEMPERATURES, *LC-NNSB*, 20 DEGREE-DAYS. AVERAGE TEMPERATURE: *On* 461; *Off* 188.

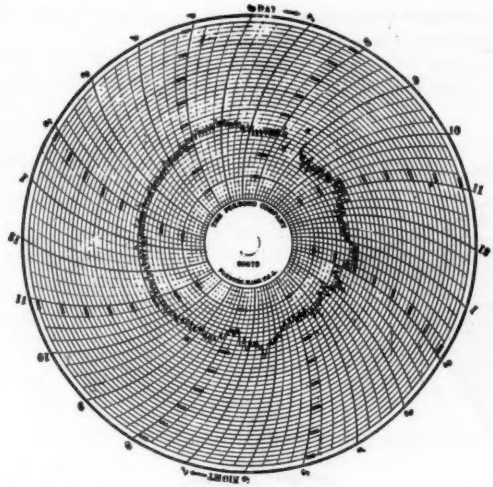

FIG. 13. RESIDENCE *A* STACK TEMPERATURES, *SC-NNSB,* 20 DEGREE-DAYS. AVERAGE TEMPERATURE: *On* 264; *Off* 264.

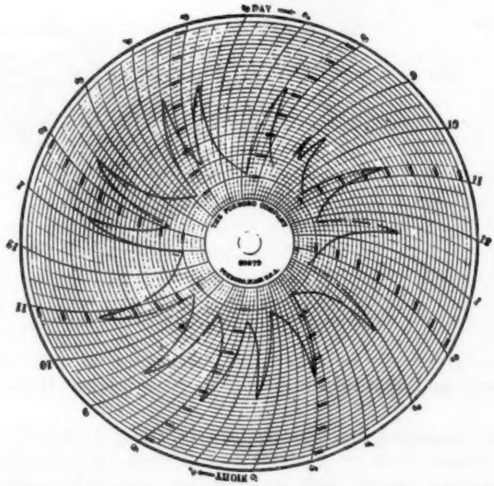

FIG. 14. RESIDENCE *A* STACK TEMPERATURES, *LC/SC-NNSB,* 20 DEGREE-DAYS. AVERAGE TEMPERATURE: *On* 379; *Off* 208.

tion with night set-back conditions than with no night set-back, due to the fact that the *off* period loss is less with night set-back.

A one week chart (Fig. 17) indicates outside weather conditions. Charts similar to these were used to determine the number of degree-days.

CONCLUSIONS

The following conclusions seem to be justified from the results obtained:

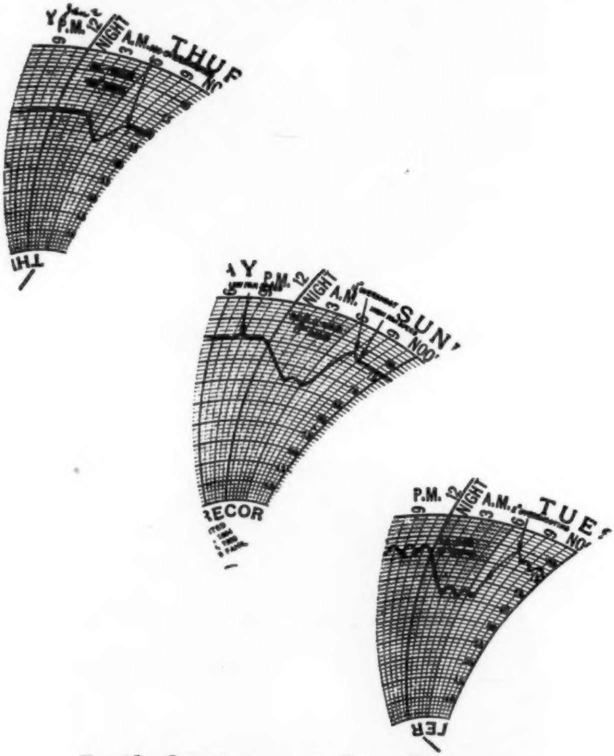

FIG. 15. OVERSHOOTING AND INSIDE TEMPERATURE.

1. There was practically no difference between the economy of *Close Temperature* control and that obtainable with conventional *Wide Temperature* operation.

2. From the standpoint of comfort, *Line* or *Close Temperature* control obtainable with either short or medium cycling was more desirable than conventional thermostat control with its attendant greater temperature variation.

3. The average flue loss for short cycle operation was less than medium cycle operation, which was less than long cycle operation. The average *off*

period loss for short cycle operation was greater than that for medium cycle operation which was slightly greater than that for long cycle operation.

4. It is apparent that if designers of burners and boilers, as well as control equipment, concentrate their efforts towards the elimination of *off* period loss, short cycle operation should be the most economical.

5. From the viewpoint of economy there was little difference between a 30 in. thermostat mounting and a 5 ft mounting, but from the standpoint of human comfort the 30 in. mounting is more desirable.

6. Although unconsumed hydrogen and hydro-carbon loss was slightly greater for short cycle than for long cycle operation, there was little difference between the two types and the little difference which existed was more than offset by the

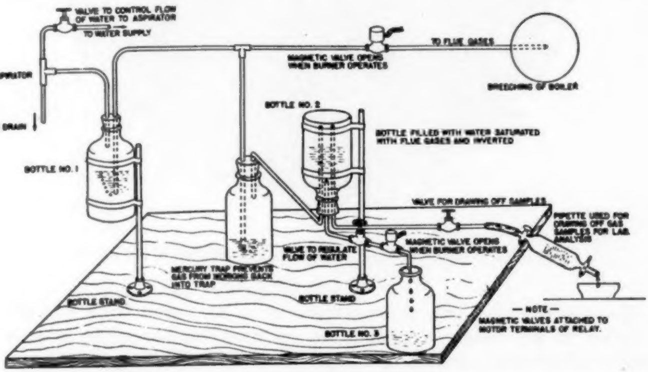

Fig. 16. Continuous Aspiration Method of Collecting Flue Gas Samples During Burner Operating Period.

lower sensible heat loss during the *on* period, because of lower temperatures resulting from short cycle operation.

7. At the two installations investigated night set-back of approximately 6 F resulted in a fuel saving dependent upon the type of thermostat operation and the degree-days. In general the saving was between 8 and 10 per cent. This is considered higher than may normally be expected.

8. Morning overshooting was least with the artificially heated thermostats and was to a minor extent dependent on degree-days.

9. It was possible to obtain short, medium, or long cycling with any of the artificially heated thermostats tested.

10. The artificially heated thermostats tested were superior to the conventional type.

11. Type *G* time interval clock actuated type of thermostat compared favorably with the artificially heated type.

12. Because of decreasing time *off* as outside temperature conditions became colder, greater economy of fuel consumption resulted, and therefore oil consumption per degree-day was not a constant.

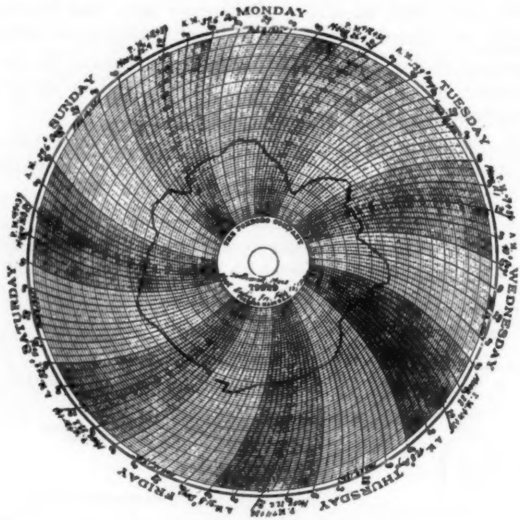

FIG. 17. RESIDENCE *A* OUTSIDE TEMPERATURE CHART.

13. Perhaps the most important conclusion to be drawn from this investigation relates to the inherent advantages of short as compared to long cycle operation. The tests indicated that short and medium cycling resulted in greater room comfort conditions. Also because of lower flue gas losses, short cycle operation would give the ultimate in economy of operation if present designs permitted more complete control of *off* period losses. Medium cycling (*Close Temperature* control within about 1 F room temperature variation) will result in but very little increased oil consumption and yet affords the maximum possible in room comfort.

ACKNOWLEDGMENT

The author is indebted to J. Smith, Research Engineer of the Gilbert & Barker Manufacturing Company, Springfield, Mass., for his able assistance in performing most of the field work encompassed by this report as well as his advice in its compilation. The author also wishes to give credit to LaVerne Lyon, Research Engineer at the Penn Electric Switch Company, for his painstaking care in the preparation of this manuscript.

APPENDIX

A group of sample calculations are included in this paper under Appendix *A*. The method of flue gas analysis used is discussed in Appendix *B*, and Appendix *C* gives the average fuel analysis used in this investigation.

APPENDIX A

1. *Off*-Period Loss

Assumptions:

Height of Stack = 30 ft

Diameter = 8 in. iron pipe

Draft vs Degree-Day Curve

a. 20 Degree-Day, Temperature in Stack = 273 F *sc–NNSB*,

Draft = 0.015 in. H_2O, ρ = (0.00086) 62.3 (from graph)

$$\Delta\rho\dagger = \frac{2fL\rho U^2}{gd}$$

$$U^2 = \frac{gd\Delta\rho}{2fL\rho}$$

$$U = \sqrt{\frac{gd\Delta\rho}{2fL\rho}}$$

Where:

$\Delta\rho$ = Draft (inches H_2O)

U = Velocity (feet per second)

ρ = Density (pound per cubic foot)

d = Diameter (feet)

D = Diameter (inches)

g = 32.2

$$\frac{s}{z} = \frac{\text{Specific gravity}}{\text{Relative viscosity}} = 0.037$$

f = Friction

Assume f = 0.0015

$$U = \sqrt{\frac{(0.015)\ (32.2)\ (0.67)}{2\ (0.0015)\ (0.00086)\ (62.3)\ (30)}}$$

$$= \sqrt{67.3}$$

= 8.21 ft per sec.

Check friction assumption.

$$(DU)\left(\frac{s}{z}\right) = (8)\ (8.21)\ (0.037) = 2.43$$

(From graph: f = 0.0016)

b. Volume per sec = (U) (Area) = (8.21) (πr^2) = (8.21) (3.14) $(4/12)^2$ = 2.89 cu ft per second.

c. Volume per day = (2.89) (Time Burner Off) = (2.89) (73,600 seconds) (from stack temp. chart) = 213,000 cu ft per day.

d. Weight per day = Density × Volume = (0.00086) (62.3) (213,000) = 11,450 lb per day.

e. Btu per day = (Temp. Increase) (Specific Heat) (pound) = (273−50) (0.25) (11,450) = 640,000 Btu per day.

f. Btu lost per Degree-Day $= \frac{\text{Btu per day}}{\text{Degree-Days}} = \frac{640{,}000}{20} = 32{,}000$ Btu.

2. *On*-Period Loss

Data:

Average Stack Temperature = 192 F

a. Dry flue gas loss.

(1) Weight dry flue gas per pound of fuel

$$W = \frac{11\ CO_2 + 8\ O_2 + 7\ (N_2 + CO)}{3\ (CO_2 + CO)} \times C + \frac{S}{1.83}$$

$$= \frac{11\ (5.8) + 8\ (10.1) + 7\ (82.24 + 0.1)}{3\ (5.8 + 0.1)} \times 0.861 + \frac{0.0006}{1.833}$$

= 35 lb per pound of fuel.

† From: Haslam and Russell, *Fuels and Their Combustion.*

(2) Heat lost per pound of fuel

$= W$ (Specific Heat) (Stack Temp – Temp of entering air) =

(35) (0.25) (390 – 50) = 2,980 Btu per pound.

b. Loss due to hydrogen in fuel.

$L_h = H \times 9\,[(212 - T_f) + 970.4 + 0.48\,(T_s - 212)]$

Where:

H = per cent H in fuel by weight.

T_f = Temperature of entering fuel.

T_s = Temperature of flue gases.

$L_h = (0.139)\ (9)\ [(212 - 50) + 970.4 + 0.48\ (390 - 212)]$

= 1520 Btu per pound fuel.

c. Loss due to incomplete combustion of carbon.

$$LCO = \frac{CO}{CO_2 + CO} \times C \times 10{,}160$$

$$= \frac{0.1}{5.8 + 0.1} \times 0.139 \times 10{,}160$$

= 17.5 Btu per pound fuel.

d. Loss due to Methane, Ethane, Hydrogen in flue gas.

(1) Change per cent by volume to per cent by weight.

Flue Gas Analysis	By Volume Per cent	Molecular Weight	$V \times M$	Per cent by Weight	$\frac{V \times M}{2887.8}$
CO_2	5.8	44	255.0	..	..
O_2	10.1	32	323.0	..	..
H_2	0.5	2	1.0	$-\frac{1}{2887.8}$	= 0.000345
CO	0.1	28	2.8	..	..
C_2H_6	0.13	30	3.9	$-\frac{3.9}{2887.8}$	= 0.00135
CH_4	0.13	16	2.1	$-\frac{2.1}{2887.8}$	= 0.00072
N_2	82.24	28	2300.0	..	..
	100.00		2887.8		

$$(2)\quad \frac{(\%\ \text{Weight})\ (\text{Pounds flue gas})}{\text{Pounds fuel burned}} \times \frac{\text{Btu}}{\text{Pounds}} = \frac{\text{Btu lost}}{\text{Pounds fuel burned}}$$

Ethane 0.00135 × 35 × 20,400 = 9.6 Btu per pound.
Methane 0.00072 × 35 × 21,600 = 5.4 Btu per pound.
Hydrogen 0.000345 × 35 × 51,700 = 6.25 Btu per pound.

3. Chimney Draft

Draft equals difference in pressure of two equal columns of air, one at stack temperature and the other at outside temperature.

$$\text{Pressure} = \frac{\text{Weight}}{\text{Area}}$$

$$\text{Weight} = \text{Volume} \times \text{Density}$$
$$Wt = V\rho_1$$
$$Wt_2 = V\rho_2$$

$$\frac{Wt_1 - Wt_2}{A} = \frac{V\rho_1 - V\rho_2}{A}$$

$$= \frac{V}{A}(\rho_1 - \rho_2)$$

$$= H(\rho_1 - \rho_2)$$

Where: H = height in feet

At 20 Deg Day *sc—NNSB* Ave Stack Temperature = 273 F *off* period an l $H\rho$ = draft in pound per square foot

Outside Temperature = 45 F

$$\rho_1 = 0.00122$$
$$\rho_2 = 0.00085$$
$$\rho_1 - \rho_2 = 0.00037$$

$$H\rho = (30)\,(62.3)\,(0.00037) = 0.693 \text{ lb per square foot}$$
$$= 0.134 \text{ in. } H_2O$$

Appendix B

Flue Gas Analysis:

Carbon-dioxide, oxygen and carbon-monoxide were determined in the usual manner using an Orsat apparatus. The carbon-monoxide was then checked by oxidation—slow combustion in oxygen in the presence of a glowing platinum coil. The carbon-dioxide was absorbed with potassium hydroxide.

Hydrogen was similarly found by slow combustion in oxygen in the presence of a glowing platinum coil. Since the water formed was negligible, the hydrogen was equal to the contraction in volume caused by its oxidation.

Methane and Ethane were also determined by slow combustion in oxygen in the presence of a glowing platinum coil. Both contraction in volume due to combustion and volume of carbon-dioxide formed was determined as Methane and Ethane were present together.

Appendix C

(Table 1, *American Oil Burner Association*, Handbook of Oil Burning.)

Fuel Analysis	Per cent by Weight
Carbon	86.1
Hydrogen	13.9
Oxygen and Nitrogen	1.6
Sulphur	0.06
Density: 7.43 lb per gallon. Btu Content: 19,350 Btu per gallon. 144,000 Btu per pound.	

DISCUSSION

D. W. NELSON: The authors are to be complimented on the great care they used in the presentation of these data and in the collection of it. In general, I agree with the conclusions, but I believe that the paper would have more value if the curves, such as Fig. 1, had the points representing the data taken on them. We know that sun effect and wind effect have a great deal to do with the consumption of oil or any other fuel being used, and if we had those points on the diagram it would be easier to give the different curves, the proper weight.

In tests made at Wisconsin we have found a great deal of difference from one day to another, and it is necessary to take a large number of days and hope to cancel by that means the various sun and wind effects involved; so I would recommend that the original data points be added.

Whether it would result in increased oil consumption or not, we are sure that in the future we are going to have close temperature control. When people buy automatic heat, they want temperature control, and the point of whether it is going to cost 10 per cent higher will not probably be a deciding factor. Of course it would be desirable if the closer temperature control did not result in increased fuel consumption. In preliminary tests we have made we believe that the oil consumption would be higher than the amounts stated here for the closer temperature control.

It seems that one degree temperature variation is as close as should be desired as a quarter of a degree is entirely too close. When the burner is started, it takes a certain number of seconds to secure equilibrium conditions. For instance, with the type of burner used in these tests, the pressure of the oil and flow of air through the combustion space do not start instantaneously. It takes 30 seconds, perhaps, to establish equilibrium conditions.

As far as I know, oil burner men in general are of the opinion that a burner operating period of less than 10 min. will result in considerable loss in efficiency. And one degree variation, I feel, in the ordinary installation, gives somewhere near this 10 min. period of running.

By dropping the temperatures at night time, our results showed something like 8 to 9 per cent saving, with a lowering of night setting some 15 F. The saving is due to the lower temperature difference between outside and inside of the house, and it may amount to more than 10 per cent or it may amount to less, depending on the construction of the house. The better insulated a house is, and the better heat resistance a house has, the less saving there will be to having lower temperatures at night.

I think it should be mentioned, as the author has, that the efficiencies mentioned there should not be taken as representative of oil burning. In one case the efficiency was 55 per cent, and in another 65, based on stack temperature, and flue gas analysis. It is possible to have oil burning installations considerably higher in efficiency than those stated.

In the first house, the burner was delivering heat at about four times greater rate than was necessary to meet the heat losses under maximum conditions and that means the burner would only be running 25 per cent of the time, and the rest of the time would be shut down.

The same thing was true in the second house, to a little less extent, however. These two cases show what is quite general in the industry, that the oil burner men for various reasons install the burner and adjust it for a much higher capacity than is necessary to meet the heat losses. For maximum efficiency the burner should be set much closer to the actual heat requirements so that the burner must run almost continuously in the coldest weather.

In the second house where the burner setting was left as the oil burner installer had made it, the CO_2 was 5.8 per cent. The high stack temperature, which was due to the heavy rate of firing together with this low CO_2, resulted in a low efficiency.

On a survey [1] that was reported at the last semi-annual meeting, the average was found at 7.8 CO_2 and 634 F flue gas temperature, which is lower than should be accepted in the industry. The author mentions that 10 per cent CO_2 is a reasonable standard for the industry. I am glad that was mentioned, and believe that the more educational work that is done to make that a minimum standard, the better will be the position of the industry.

Many installations, such as House B referred to in this paper, are put in at a much lower efficiency than is necessary. A CO_2 content of 5.8 per cent is often the setting for a new installation, but there is no excuse for it under present-day conditions. Oil burner men should know that 10 per cent is reasonably possible and also that a flue gas temperature of not higher than 500 F is reasonable and should be insisted upon.

The special value of this paper is educational in that it presents before engineers and oil burner men the need for higher efficiency and greater care in installations.

W. A. DANIELSON: I would like to know if the fan was operated with a thermostat.

MEMBER: Were the degree days given in this paper based on the house temperature maintained over an average of 24 hours a day or on the usual standard of 65?

MR. ECKERT: Things which seem important to me in connection with this study are the gas velocity with its relation to stack temperature and the possibility of a vacuum existing in the basement which will directly affect combustion. If houses are built air tight, the fires would go out; therefore, we must provide infiltration in some manner such as through cracks in the doors and windows.

W. R. BEACH: One thing to which I would like to call your attention in regard to this close thermostat setting, or short-cycling, is the result in the power distribution lines. A recent case came to my attention where there was a group of five houses with automatic equipment of various sorts in each house, these houses all being served by one distribution transformer. The automatic equipment consisted of several refrigerators, two or three stokers, and five furnace fans, one of which was set for very frequent starting. Several voltage complaints came in from this group, and the installation of a graphic voltmeter showed that with so many automatic pieces of equipment some one of them would be starting every minute or two, causing objectionable voltage fluctuation. Therefore, I want to suggest that as you work out the automatic devices for frequent starting, it should be borne in mind that it is necessary to keep their starting currents low or to connect them on 230 volt service in order to reduce voltage fluctuation to a point which will result in satisfactory electric utility service.

B. E. SHAW: I want to thank Professor Nelson for his remarks and constructive criticism. It would have been desirable to have all of the points on Fig. 1, but I decided to eliminate them because of the mass of data obtained over a complete heating season of over 200 days. It seemed impossible to present all of the points on such a small graph, but we had to, as Professor Nelson pointed out, take the mean of the various points, hoping that sun effects and solar radiation, wind velocities, etc., would cancel out to a major extent.

I cannot quite agree with Professor Nelson that a quarter of a degree temperature variation is too close. I think constant temperature conditions would be right, providing they are not too close. It is true that the losses increase with the shorter cycling, and with some burners this might be more or less insurmountable, but that is a problem for the burner industry.

By reference to Fig. 8 and Table 5 it is apparent that equilibrium conditions are soon reached after combustion is started. The problem of high unconsumed losses referred to in practice for short cycling operation do not actually exist. The losses did increase somewhat for short cycling operation, but not to any great extent.

[1] Oil Burning in Residences, by D. W. Nelson, A. S. H. V. E. TRANSACTIONS, Vol. 41, 1935.

Professor Nelson also pointed out the high firing rate. It was necessary to use firing rates as high as shown because temperature conditions in both locations were extreme. Temperatures outside dropped to 35 F below zero in one locality and 32 F below in the other. When we consider the design temperatures were 0 to 70 F (I feel in these two localities they should have been —10 to 70 F) we can understand why it is necessary for the burner man to overfire most burner installations in order to take care of local conditions and prevent dissatisfied customers.

Professor Nelson will note that where the curves converge somewhere above 100 degree days would be the point where the burners would have operated continuously. To answer another question, the degree days were taken on the basis of 65 F and in accordance with the definition given in THE GUIDE.

The CO_2 values recorded were low, and it certainly would be advisable where possible to install units operating at 10 per cent CO_2. However, the results are comparative, and it was felt that the results obtained were more or less typical. Maybe they were too low, and the losses are somewhat high, but certainly they are comparative, in any event.

Another question referred to fan operation. During the course of investigation at Residence A the fan was operated at various speeds. Some determinations were made relative to the various effects of gravity control, slow and high speed fan operation. All of these data are referred to in this paper.

Relative to the velocity of stack gases, in Appendix A is given a complete discussion of this subject matter, and cognizance is taken of the problem of stack gas velocity.

When jobs are properly installed correct wiring diagrams should be employed so that all of the various motorized equipment does not start at the same time to prevent a surge in voltage. It is possible through stack switches to hook up fan equipment so that the fan does not start when the burner starts, but is brought on sometime later when the stack is in a hot position; also relays can be used. Undoubtedly some jobs are being installed with all of the various motors starting at one time, and will not give satisfactory operation, because of insufficient line or supply transformer capacity. In general this is a problem for the power company.

No. 1028

FUEL SAVING RESULTING FROM THE USE OF STORM WINDOWS AND DOORS

By A. P. Kratz * and S. Konzo ** (*MEMBERS*), Urbana, Ill.

Object

THE object of this investigation was to determine, under actual service over a wide range in weather conditions, the saving in fuel that could be effected by equipping a typical residence with storm windows and doors; and to compare the actual saving so effected with the saving computed from heat loss calculations employing commonly accepted values for the coefficients of heat transmission and air infiltration.

Description of Research Residence and Heating Plant

The Research Residence and heating plant have been completely described in previous publications.[1] The Research Residence, shown in Fig. 1, is a three-story structure of standard frame construction. The wall section consists of weather boarding, building paper, sheathing, 6-in. studding, wood lath, and plaster with rough sand finish. The walls are not insulated and no weatherstripping is used at the windows and doors.

The total space heated during these tests consisted of three rooms, a sun parlor, a breakfast nook, and a hallway on the first story; three rooms, a bath room, and a hallway on the second story; and two rooms, a bathroom, and a hallway on the third story. The total volume of this heated space, from which the basement was excluded, was approximately 17,540 cu ft. The calculated heat loss was approximately 137,500 Btu per hour at an indoor-outdoor temperature difference of 70 F and approximately 159,000 Btu per hour at an indoor-outdoor temperature difference of 80 F. The Research Residence is completely furnished, and during the heating season it was occupied by four people.

The heating plant consisted of a coal fired furnace used in connection with

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** Special Research Associate, Engrg. Experiment Station, University of Illinois.
[1] University of Illinois, Engrg. Experiment Station, *Bulletins Nos. 189, 246,* and *266.* Also Study of Summer Cooling in the Research Residence at the University of Illinois, by A. P. Kratz and S. Konzo, A. S. H. V. E. Transactions, Vol. 39, 1933.

Presented at the 42nd Annual Meeting of the American Society of Heating and Ventilating Engineers, Chicago, Ill., January, 1936, by A. P. Kratz.

a forced-air heating system. Three cold air returns were used which were connected into a cold-air box above the inlet to a centrifugal type of fan. The furnace was placed at the east end of the basement, and the warm air registers were served from two main trunk systems. The furnace was of the cast-iron, circular-radiator type, having a 24-in. firepot and 20-in. grate, and was equipped with a casing 42 in. in diameter.

The control [2] of the heating plant was accomplished by means of a room thermostat operating to open and close the ashpit damper and to start and stop the fan. This room thermostat was used in conjunction with two bonnet thermostats which served as high and low limit controls for the temperature of the air in the furnace bonnet. The room thermostat was located on an inside wall of the dining room at a height of 5 ft from the floor and was

FIG. 1. WARM AIR RESEARCH RESIDENCE, URBANA, ILL.

adjusted to maintain an air temperature of approximately 71 F at this level in all of the rooms of the Residence.

One series of tests was run with the Residence equipped with storm windows and door, and one series was run without such equipment. For the first mentioned series, all of the windows on the three stories of the Residence, with the exception of two small quarter-round windows in the east dormitory, were provided with tightly fitting storm sash. As shown in Fig. 2, felt stripping was placed along all four contact edges and the storm sash was clamped tightly to the window casing by means of screws. One window in each of the second story rooms was fitted with hinges at the top so that outdoor air could be admitted occasionally if desirable. The front entrance was equipped with a storm door, but the rear entrance was not. The outside door at the rear opened into a vestibule which contained the basement steps, and the kitchen door opened into this vestibule. Hence an additional storm door was not considered necessary. The areas of window and door openings, of wall surfaces, and the ratios of openings to wall surfaces are given in Table 1.

[2] Control Type IV described in the paper, Automatic Controls for Forced-Air Heating Systems, by S. Konzo and A. F. Hubbard, A. S. H. V. E. TRANSACTIONS, Vol. 40, 1934.

Method of Conducting Tests

The two series of tests selected to determine the effect of storm windows and doors comprised part of the routine test program carried on at the Research Residence. The same test methods were employed for all such tests. The fuel burned was anthracite, having a calorific value of 13,175 Btu per pound, and the controlling thermostat was adjusted to maintain a temperature of approximately 71 F at the 5-ft level in all rooms. This temperature was maintained during all of the 24 hours of the day.

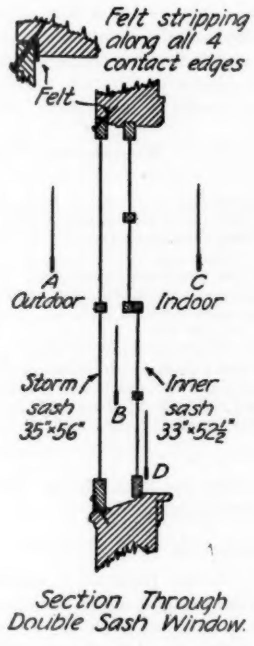

Fig. 2. Line Diagram Showing Method of Installing Storm Sash

The furnace was fired at four regular periods, namely 7:00 a.m.; 11:00 a.m.; 4:00 p.m.; and 10:00 p.m., and a record was made of the net fuel consumed during each 24-hour period. Either periodic or continuous records were made of all significant temperatures, and average daily temperatures both indoors and outdoors were obtained from these records. Each series of tests was continued over periods of sufficient length to obtain a wide range of weather conditions representative of the heating season.

RESULTS OF TESTS

Infiltration

No provision was made for the continuous introduction of outdoor air. Therefore, the only infiltration that occurred was that due to leakage around the window and door frames, leakage through the frame walls, and the influx of cold air accompanying the opening of outside doors. During ordinary conditions with normal occupancy by four persons, the slight inleakage of air that occurred was sufficient to prevent any noticeable accumulation of odors. Occasionally it was found desirable to remove the cooking odors from the kitchen by opening the door to the outdoors for a few minutes. It should be noted in this connection that the outdoor air requirements to prevent accumulation of odors are comparatively small for residence service, and it is questionable whether in the ordinary residence installation, special provision for continuous introduction of outdoor air would be necessary. Any such special ventilation

TABLE 1. DATA ON WINDOW AND WALL SURFACES

1	Number of window openings	50
2	Number of windows equipped with storm sash	48
3	Number of door openings to outdoors	2
4	Number of storm doors	1
5	Area of exposed window openings, square feet	525
6	Area of windows equipped with storm sash, square feet	522
7	Area of exposed door, square feet	24.5
8	Area of double door, square feet	24.5
9	Gross area of exposed structure, square feet	3004.5
10	Net area of exposed wall (windows and doors excluded), square feet	2455
11	Ratio of storm windows and doors to total exposed openings, per cent	99.4
12	Ratio of total exposed openings to gross wall, per cent	18.3
13	Ratio of total exposed openings to net wall, per cent	22.4

would, of course, partly offset the heat saving that could be accomplished by the application of tightly fitting storm sash.

Comparison of Fuel Consumption

The data on the amount of fuel required to operate the Research Residence with and without storm windows and doors are shown in Fig. 3, in which the daily fuel consumption, in pounds of coal, is plotted against the difference in temperature between the indoors and outdoors. The deviations of the points from the mean curve are primarily caused by differences in daily fuel consumption brought about by variable wind and sun effects, which cannot be represented on a curve in which the abscissa is temperature difference alone. However, the deviations resulting from wind and sun effects tend to compensate when a number of tests are conducted with the same indoor-outdoor temperature difference, and the mean curve becomes representative of the actual fuel consumption when considered from the standpoint of the season as a whole.

The curves in Fig. 3 show that the average daily amount of fuel required to heat the Residence when the outdoor temperature was 40 F, which corresponds closely to the mean seasonal temperature in Urbana, Ill., was 100 lb when storm doors and windows were not used and 81 lb when storm doors and windows were used. This represents a saving in fuel consumption of 19 per cent attributable to storm doors and windows. The saving in milder

weather was somewhat less, but in severer weather the saving increased to a value of 21 per cent shown for zero weather, or at an indoor-outdoor temperature difference of 70 F. The results therefore indicate that a saving of approximately 20 per cent in the seasonal fuel consumption could be reasonably attributed to the installation of storm doors and windows on the Research Residence.

Calculated Reduction in Heat Losses

The calculated reduction in heat losses was obtained by computing the heat losses from the structure both with and without storm windows and doors. The difference between the two calculated values would be accounted for in the items involving infiltration and heat transmission through windows and doors.

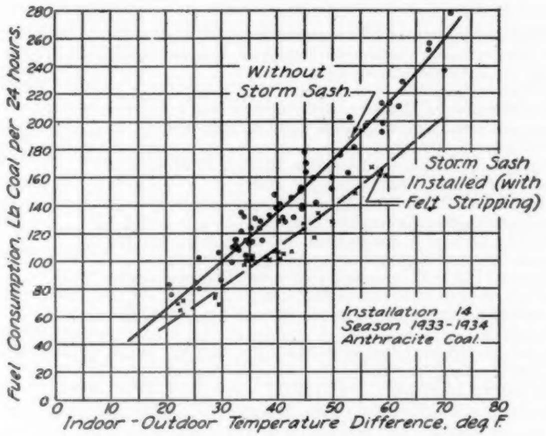

FIG. 3. FUEL CONSUMPTION WITH AND WITHOUT STORM SASH.

These items were therefore calculated for a zero day, or for an indoor-outdoor temperature difference of 70 F, and added to the basic heat loss of 73,770 Btu per hour, which took place through walls, ceilings, floors and all parts of the structure exclusive of the exposed windows and the front door. The data used for the calculation of the infiltration and heat transmission losses through the windows and door are given in Table 2, for which the coefficients were obtained from THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS GUIDE 1935. The infiltration coefficients given in Item 8 were based on a well fitted window having 5/64 in. crack and 1/32 in. clearance. A frame leakage of 10.8 cu ft per hour obtained from THE A. S. H. V. E. GUIDE 1935, Table 2, page 123, was added to the leakage for windows with and without storm sash, read at a wind velocity of 15 mph from the curves on page 125 of THE A. S. H. V. E. GUIDE. These totals were then reduced 20 per cent to allow for building up of pressure within the building and the results were multiplied by 0.075 and 0.24 to obtain the coefficients in terms of Btu per lineal foot of crack per degree Fahrenheit per hour. The infiltra-

tion coefficient for the unprotected door was obtained by using the value of the leakage for a poorly fitted window, as recommended. No data were available for the leakage around a storm door, but since the one used closed against felt strips, it was regarded as a weatherstripped door, and the leakage was assumed as one half of that for a door without weatherstripping. The front door was located in a recess which formed a shallow vestibule when the storm door was installed. The heat loss under these conditions was calculated by using the same coefficient of heat transmission for the two cases, but regarding

TABLE 2. HEAT LOSS DATA, BASED ON 70 F TEMPERATURE DIFFERENCE

No.	ITEM	WITHOUT STORM SASH	WITH STORM SASH
1	Heat loss through walls, floors, and ceilings, Btu per hour	73,770.0	73,770.0
2	Lineal feet of crack around windows (one-half of total)	356.0	356.0
3	Lineal feet of crack around door	21.0	21.0
4	Total area of windows, square feet	525.0	525.0
5	Area of windows with storm sash, square feet	...	522.0
6	Area of windows without storm sash, square feet	525.0	3.0
7	Area of door, square feet	24.5	24.5
8	Infiltration coefficient for windows, Btu per lineal foot of crack per degree Fahrenheit per hour	0.74	0.34
9	Infiltration coefficient for door, Btu per lineal foot of crack per degree Fahrenheit per hour	2.00	1.00
10	Coefficient of heat transmission for windows, Btu per square foot per degree Fahrenheit per hour	1.13	0.45
11	Coefficient of heat transmission for door, Btu per square foot per degree Fahrenheit per hour	0.52	0.52
12	Calculated heat loss through doors and windows, Btu per hour	63,800.0	27,070.0
13	Total calculated heat loss from building, Btu per hour	137,570.0	100,840.0
14	Calculated saving, per cent	...	26.7
15	Actual saving from tests, per cent	...	21.0

the door exposed to the outdoor temperature in one case and the mean between the indoor and outdoor temperature in the other.

From Table 2 it may be observed that the calculated heat loss was 137,570 Btu per hour for the building not equipped with storm windows and doors, and 100,840 Btu per hour for the building equipped with storm windows and door. This represented a computed saving of 26.7 per cent as compared with the actual saving of 21 per cent shown at an indoor-outdoor temperature difference of 70 F by the test curves in Fig. 3. This may be regarded as very close agreement considering the uncertainties which necessarily accompany the computation of infiltration losses, and several explanations may be offered to account for the fact that the apparent saving was somewhat greater than the actual. The published coefficients are based on laboratory tests under controlled and readily determinable conditions. In applying these coefficients, the question as to how nearly the actual conditions approximate those stated for the laboratory tests is largely a matter of estimation and judgment.

In the case of the storm windows, the storm sash were positively secured against felt strips. This condition can be readily duplicated and it is probable that coefficients determined from laboratory tests applied reasonably well to the actual installation. Furthermore, the leakage is small as compared with the

total heat loss from the building. In the case of the unprotected windows the infiltration depends on the width of crack and the clearance around the frames, both of which are difficult to estimate or measure accurately. This leakage is relatively larger than that around the storm sash, and if it were overestimated it would indicate an apparent saving that would be greater than the actual. It is possible, therefore, that the unprotected windows at the Research Residence were tighter than allowed for in the estimate, thus accounting for the larger apparent saving indicated by the computed results.

The calculations for heat losses do not take into consideration the admission of cold air through doors at the time of entrance. If this loss remained constant in the two installations, it would be a relatively greater proportion of the total heat loss when the building was equipped with storm doors and windows than it would be when the building was not so equipped. Hence neglecting this item in the computations would tend to increase the apparent saving as compared with the actual saving.

The calculated heat losses were based on a wind velocity of 15 mph in order to maintain consistency with the values of the heat transmission coefficients selected from THE A. S. H. V. E. GUIDE, whereas the actual average wind velocity in Urbana, Ill., in zero weather is somewhat less than this. The wind has a greater effect on unprotected windows than on those protected by storm sash, and at lower velocities the differences in infiltration would not be as great as they would be at higher velocities. This would also tend to indicate an apparent saving greater than the actual.

Fuel Saving as Affected by Features of Construction.

It is obvious that the percentage of reduction in fuel consumption with and without the use of storm doors and windows is dependent on the nature of the wall construction and the ratio of exposed window surfaces to the net wall surface. For a given room or house, storm windows and doors will effect a larger percentage saving when the wall is well heat-insulated than when it is not. Also, for two rooms of the same size having the same wall construction, but different ratios of window surface to net wall surface, the percentage of fuel saving will be greater when storm windows are applied to the room which has the larger ratio of exposed window surface to net wall surface, than it will be when the storm windows are applied to the room having the smaller ratio.

In this connection, tests of storm windows and doors made under laboratory conditions are of some interest. These tests were made in the room heating testing plant in the Mechanical Engineering Laboratory at the University of Illinois. This plant [8] consisted of two test rooms, each having two wall exposures of frame construction, erected inside of a large insulated enclosure. The test rooms were heated with steam radiators, and, by means of refrigerating coils in the large enclosure, the two walls of the test rooms could be exposed to any desired air temperature. For these tests the temperature on the outside of the exposed walls was maintained at 0 F, and the heat loss from the test room was measured by the steam condensation from the radiator. The test room was 9 ft by 11 ft with a 9-ft ceiling, and had two double-hung windows each 2 ft 6 in. by 4 ft 6 in., and an exposed door 3 ft by 7 ft. The ratio of the window area to net wall area was 16.5 per cent and the ratio of

[8] University of Illinois, Engrg. Experiment Station, *Bulletin No. 223*, Chapter II, pp. 11-17 and Chapter IX, pp. 62-66.

the sum of the areas of the windows and door to the net wall area was 31.9 per cent. In these tests the steam condensation required to heat the room to 70 F at the 5-ft level was reduced 11.0 per cent when storm windows alone were used, and 31.0 per cent when both storm windows and storm door were installed. The results obtained in the Research Residence, in which the wall construction was similar to that of the laboratory test room, were consistent with those obtained in the laboratory tests. The actual fuel saving was 21.0 per cent, which was intermediate between the reductions in steam condensation of 11.0 per cent and 31.0 per cent shown in the laboratory tests with storm windows alone and with the storm windows and door respectively; while the ratio of the area of openings to that of the net wall was 22.4 per cent in the Research Residence, which was intermediate between the corresponding ratios of 16.5 per cent and 31.9 per cent for the windows alone and for the windows and door in the laboratory test room. The fact that the per cent saving was approximately the same as the ratio of the area of openings to the net wall area in each case is probably only a coincidence, but it is evident that the potential saving increases as the ratio of openings to net wall becomes greater

Incidental Data

In addition to fuel saving, other advantages incident to the operation with storm sash became evident. The tightly fitting storm sash practically eliminated the entrance of soot, which in the case of the unprotected windows sifted in and collected in sufficient quantity on the white window stools as to make daily cleaning necessary.

The use of storm windows enabled the maintenance of higher indoor relative humidities without condensation on the windows. Observations [4] made at the Research Residence, simultaneously on windows not equipped with storm sash and on two windows provided with storm sash, during a period when the relative humidity indoors was being rapidly increased with a constant outdoor temperature of 26 F, proved that condensation started to appear on the unprotected windows when the relative humidity reached a value of 32 per cent. No condensation appeared on the two windows equipped with storm sash. This agrees very closely with computed curves indicating that, with 40 per cent indoor relative humidity, condensation appears on unprotected windows when the outdoor temperature drops to 35 F while with storm sash the outdoor temperature must drop to 0 F before condensation appears. Operation during a whole season with all of the windows protected with storm sash, however, indicated that if there is any appreciable leakage around the inner sash, some condensation will deposit on the glass in the storm windows in extremely cold weather when high relative humidities are maintained indoors.

In addition to reducing the heat loss from the building, storm windows were also effective in reducing the downward draft of cold air usually present with unprotected windows.[5] This is shown in Fig. 4 from which it may be observed that the temperature of the current of air coming down over the windows, as measured at *D* (see also Fig. 2) was approximately 3 F greater for the windows equipped with storm sash than it was for those not so equipped.

The immediate effect of this reduction in draft was an increase in the air temperature in the living zone of the room, particularly near the floor. The

[4] University of Illinois, Engrg. Experiment Station, *Bulletin No. 266,* Chapter XI, pp. 115-121.
[5] Loc. Cit. See Note 4.

observed room temperature gradients in the dining room are shown in the left hand portion of Fig. 5 for the case (*B*) in which the windows were not protected and for the case (*C*) in which storm windows were used. It may be noted that the air temperature gradient from the 5-ft level to the ceiling was not affected by the installation of storm windows.

With a fixed setting of the controlling thermostat, there was a marked reduction in the total time of fan operation in the forced-air system, accompanying the reduction in fuel saving when the storm windows were installed. For an average outdoor temperature of 25 F, as illustrated in the right half of Fig. 5, the ratio of the time of fan operation to the total time was approxi-

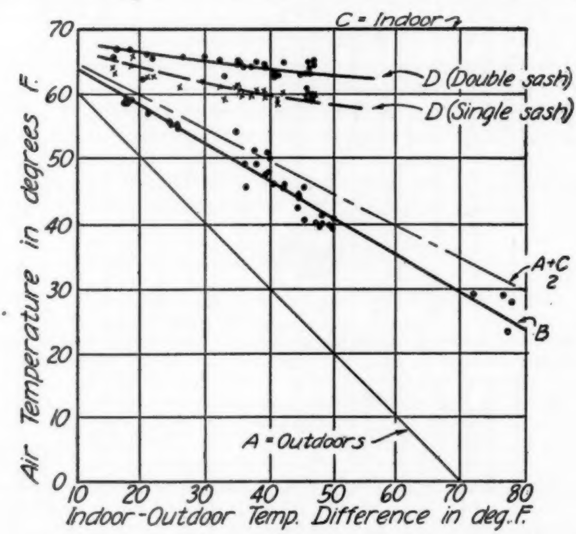

FIG. 4. AIR TEMPERATURES IN CONNECTION WITH STORM SASH (THERMOMETER LOCATIONS INDICATED BY *A*, *B*, *C*, AND *D*. SEE FIG. 2).

mately 40 per cent without storm windows and 30 per cent with storm windows. Since the percentage of time of fan operation compared to total time varies with different plants and with different settings of the controlling thermostats, the numerical values of these percentages cannot be regarded as applying to any plant except the one under consideration. They do, however, serve to illustrate the comparative performance of the plant with and without storm windows. The advantages cited with the use of storm windows emphasize the importance of making adequate provision to reduce the heat losses at what may be regarded as the most vulnerable part of the structure from the standpoint of the heating installation, namely the doors and windows. The problem of heating a room becomes greatly simplified when the ordinary unprotected windows are replaced by adequately protected windows, either in the form of weatherstripping, double-glass, or tightly fitted storm sash, although

weatherstripping or double-glass alone are probably not as effective as tightly fitting storm windows.

CONCLUSIONS

The following conclusions may be drawn from the results of these tests:

(1) A seasonal fuel saving of approximately 20 per cent may be obtained

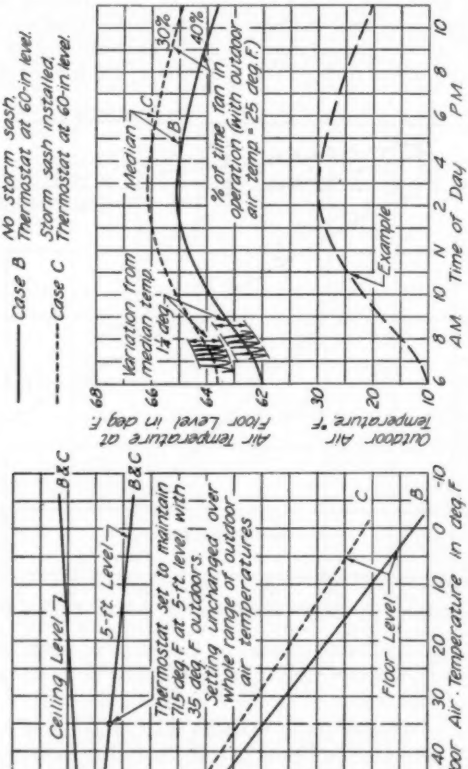

FIG. 5. ROOM TEMPERATURE GRADIENTS OBTAINED WITH AND WITHOUT USE OF STORM SASH WINDOWS. ROOM THERMOSTAT LOCATED 60 IN. ABOVE FLOOR.

by equipping a frame building similar to the Research Residence with storm windows and storm doors.

(2) A reasonable agreement can be obtained between the fuel saving effected by storm windows and doors as computed from the calculated heat losses and the actual saving as determined by tests. The computed probable saving tends to be higher than that actually demonstrated by tests.

(3) For a given type of wall construction, the fuel saving effected by storm windows and doors is dependent on the ratio of the area of the windows and doors to the net area of the walls. The potential saving increases as this ratio becomes greater.

(4) Tightly fitting storm sash practically eliminates the entrance of objectionable amounts of soot.

(5) Storm windows make possible the maintenance of higher indoor relative humidities without condensation appearing on the glass.

(6) The use of storm windows reduces the draft of cold air down the windows and increases the temperature of the air near the floor of the room.

ACKNOWLEDGMENTS

The results presented in this paper were obtained in connection with an investigation of warm-air furnaces and heating systems which is being conducted by the Engineering Experiment Station of the University of Illinois in cooperation with the *National Warm Air Heating and Air Conditioning Association*. The basic data were obtained at the Research Residence[6] (Fig. 1) and the results will ultimately comprise part of a bulletin of the Engineering Experiment Station of which Dean M. L. Enger is the Director.

DISCUSSION

MR. HAAS: I am interested to know more in detail about the damper control and whether or not the fire was hand-fired.

F. B. ROWLEY: I would like to ask Professor Kratz whether or not he made calculations to determine the heat losses and if so how closely the calculated results agreed with the test results.

B. E. SHAW: I would like to ask how many people slept in the dormitory and whether the night temperature was dropped; if so, how much? Also, what type of thermostat was employed and what degree of regulation was obtained at the thermostat location on a sensitive recorder?

MEMBER: Could the arrangement of the cold air returns make a difference in the floor temperature?

MINNESOTA MEMBER: I live in Minnesota and we always employ storm windows, but they are never applied the way they were applied in the research residence. We are interested to know whether the storm windows as applied in the test were applied the same way in the laboratory, to prepare the theoretical charts, and we also would like to know whether any further tests will be made with storm windows as actually used in practice, as the difference should be very great as to the benefit of floor temperatures and surface temperatures on the inside of the glass.

W. A. DANIELSON: I would like to ask if the inside windows were weatherstripped and what was the standard wall construction?

A. P. KRATZ: I confess I haven't got all of this discussion, but I will try to answer some of it, and if I overlook anything, it will not be intentional. In reply to Mr. Haas, with reference to our damper control, in the first place the furnace was hand fired, using anthracite. The damper was controlled from the room thermostat, but two thermostats in the bonnet of the furnace acted in conjunction with the damper control. That is, if the bonnet temperature was too high when the room thermostat called for heat, the fan started, but the damper was not opened immediately.

[6] The Research Residence in Urbana, Ill., was built, furnished and completely equipped especially for research work in warm-air heating by the *National Warm Air Heating and Air Conditioning Association* in December, 1924.

On the other hand, if the bonnet temperature was too low, in order to keep from blowing cold air into the room, the damper opened, but the fan did not start until the temperature in the bonnet rose to a reasonable extent. My recollection is that we had the bonnet temperature set betwen limits of 130 and 170 F.

I am not sure that I grasped all of the points Professor Rowley brought out, but I think he is correct in that you find a variation in the coefficient with the mean temperature of the wall, and that as the mean temperature gets lower, the effectiveness of the wall as an insulator increases, and while I had not checked it on these tests, I would not be surprised if they do show that effect. I think he also asked how closely the actual and calculated values agreed. My recollection is that the calculated and actual heat losses agreed within about 10 per cent.

Mr. Shaw asked how many people were in the dormitory. There were two people, that is, the dormitory rooms were occupied, each room, by one person, and so there was one person sleeping in each one of those two dormitory rooms. In addition to that, there were two people sleeping on the second floor, and, I believe they did not open their windows at night. I think we made it conditional during that time not to open the windows at night, and it was not necessary to get ventilation in the house. The thermostat regulated within about a degree; from a degree to a degree and a half.

Some one asked whether any difference would be shown by a change in the location of the cold air grilles. We did not change the location of the cold air grilles during the time we were running these tests, so I cannot answer that very definitely. With the forced air system it might well be that different locations of the cold air grilles would give slightly different floor temperatures at the center of the room where our temperatures were taken.

Some one asked whether the storm sash was applied for the test results in the same manner as that used for the laboratory results. Professor Larson can correct me if I am wrong in this. The coefficients for storm sash were determined under almost the same conditions as used in the tests. They were determined for a tight storm sash clamped against a felt strip.

G. L. LARSON: Not the felt strip.

PROFESSOR KRATZ: But they were clamped tight.

PROFESSOR LARSON: Yes.

PROFESSOR KRATZ: We had not considered running any more tests on that at present, unless there is a demand for it.

Colonel Danielson asked about insulation, I believe, and weatherstrips. No weatherstrips were used on the inside sash. The house has never been weatherstripped, and there is no insulation in the house with the exception of the ceiling above the northwest bedroom, where an inch of insulating blanket has been nailed on top of the joists in the attic space above that bedroom. Outside of that, the house is not insulated, and the only deviation from standard frame construction with weather boarding, sheathing, studding, and wood lath and plaster, is that 6-in. studding was used throughout instead of 4-in.

COLONEL DANIELSON: Is there paper in the walls?

PROFESSOR KRATZ: Yes, there is building paper between the sheathing and siding. The inside walls were painted and not papered.

W. C. RANDALL: When storm windows are used, a saving is effected such as the opening of a storm window at night to get ventilation, or the opening and closing of a door. It would seem that, after all, the safe thing to do is take the percentage saving that is computed in this particular proposition, and then use 75 per cent of that as being in the general range of what the saving would be in the fuel.

It is quite possible that the computed value of *U* for single windows is in error, and if that value were lowered, the difference between single and double windows, as computed, would be closer to the results experimentally obtained.

PROFESSOR KRATZ: "Was the temperature lowered at night?" was another question. No, during these tests the temperature was not lowered at night.

No. 1029

PERFORMANCE OF FIN-TUBE UNITS FOR AIR HEATING, COOLING AND DEHUMIDIFYING

By G. L. Tuve * (*MEMBER*)
Cleveland, Ohio

This paper is the result of research sponsored by the American Society of Heating and Ventilating Engineers in cooperation with Case School of Applied Science

Introduction and Summary

SINCE presentation of the last report on this project,[1] the work has consisted of about 250 one-hour test runs, and correlation of the results with data published by other investigators and with the performance tables issued by manufacturers of fin-tubing.

General conclusions are that, while there is now a fair knowledge of the laws of performance of fin-tubes, the data at present available to engineers are in some respects inconsistent. More study and research is necessary, especially in the field of dehumidification, in order to establish a satisfactory basis for uniform methods of rating these units. It would seem desirable for the A.S.H.V.E. to establish a simple and uniform method for testing any given design of extended surface unit, and to specify certain features in the developing of complete performance tables from such tests.

One simple method of establishing the ratings is suggested in this paper and the methods of applying it are discussed. Numerical results on the performance of a large number of coils are given, together with information on how the capacity of a coil changes when the operating conditions are changed. A table of typical performance values is recommended for use in estimating or checking the capacity of a fin-tube coil for any ordinary air-conditioning service.

Is There a Simple Way of Stating the Performance of a Fin-Tube Coil?

In the final practical application involving the selection of a coil for a given service, a complete performance table should be available for each working

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[1] Heat Transfer from Direct and Extended Surfaces with Forced Air Circulation, G. L. Tuve and C. A. McKeeman, A. S. H. V. E. Transactions, Vol. 40, 1934.

Presented at the 42nd Annual Meeting of the American Society of Heating and Ventilating Engineers, Chicago, Ill., January, 1936.

fluid, *i.e.,* steam, cold water, dichlorodifluoromethane, etc. Each table should show the change in performance due to changes in the velocity, temperature and humidity of the entering air, and due to changes in the number and arrangement of the tube elements. For hot or cold water circulation, the water velocity must also be taken into account.

But is there not some simple way of stating the basic performance factors for a given type of extended surface, so that the effect of changes in operating conditions may be readily calculated? Common practice among engineers answers this question by the almost universal use of the *overall coefficient* as the basis for general heat transfer calculations. Specialists in heat transfer analysis or testing must use the individual or surface coefficients in their calculations, but most engineers who select or specify heat transfer units will prefer to work from the overall coefficients. Such a viewpoint is justified in the case of multiple banks of fin-tubes, because of the additional difficulty of separating out the performance of the individual rows or banks, and because of the complication of temperature gradients in the fins.[2] Moreover, it has been proven by experiment that under almost all conditions the simple exponential equation applies to the relation between the overall coefficient and the air velocity, *i.e.,* $U = \text{const.} \times V^n$. It will be noted that this is the equation of a straight line on logarithmic coordinates. The value of the coefficient for any and all air velocities can, therefore, be obtained if only one point and the slope of this line are known.

Makers and users of extended-surface units have gradually adopted the practice of using this straight-line relation on the logarithmic graph of capacity against air velocity, so that most performance tables are now built on this basis. But this relation could to advantage be more generally used by engineers, and it could well be used as a basis for establishing a standard method for testing and rating fin-tube coils.

To show the practical possibilities of this simple point-slope method of expressing fin-coil performance, the test results presented in this paper and the performance data quoted from other sources will be shown by this method. For all types and arrangements of coils, the performance at 500 fpm air velocity through the face area will be used as the basic point or reference value (symbol U_{500}) and the slope will be given (symbol n) for the straight line which shall be drawn through this point when it has been located on logarithmic coordinates.

FACTORS AFFECTING FIN-COIL PERFORMANCE

Viewing the problem from this angle of fixing a point and then drawing a line of a given slope through this point, attention may be confined to two questions:

1. What factors determine the capacity of a fin-tube unit at the *standard* face velocity of 500 fpm.[3]

2. What factors determine the slope of the straight line (on logarithmic

[2] Basic heat transfer equations for fin-tubes were given and discussed in the former report on this research (see footnote [1]).

[3] This *standard* velocity is arbitrarily selected as representing a common value in practice. Any other value might be used if desired.

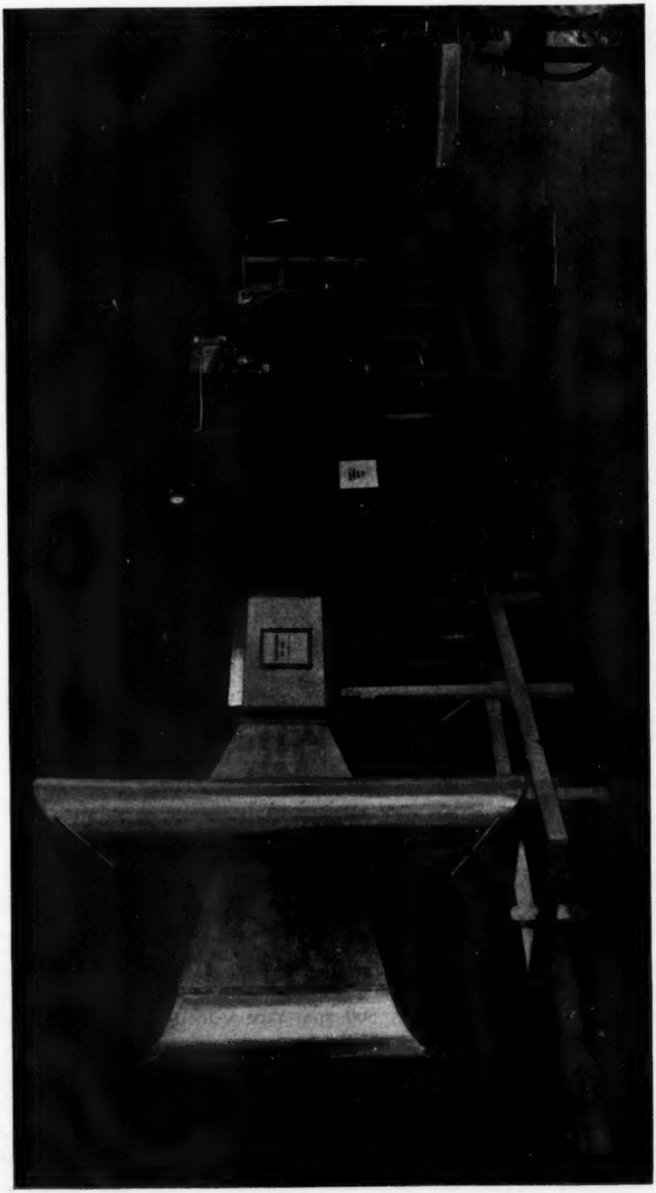

FIG. 1. GENERAL ARRANGEMENT OF TEST DUCT SHOWING A SINGLE-ROW COIL UNDER TEST AND ELECTRIC BOILER WITH AUTOMATIC CONTROL.

coordinates) showing the relation between coil capacity and face velocity of the air?

FIN-COIL PERFORMANCE TABLES

As already suggested, the *overall coefficient* (U or K), is the most convenient measure of the capacity of a fin-tube unit. This *specific capacity* is expressed in Btu per hour per square foot of air-side surface area per degree

TABLE 1. PERFORMANCE OF 25 TYPES OF COPPER CIRCULAR-FIN-TUBE UNITS
(*Dipped metal bond or integral fins*)

No.	Data Source	Working Fluid	No. of Rows	In-Line or Staggered	Tube Diam. In.	O.D. Fins In.	Fins per In.	Free Area %	U at 500 fpm	n Slope of Line
Air Heating Service										
1	Reported[a]	Steam, 2–100 lb	4	Stag.	0.375	0.87	6	57	13.0	0.62
2	Research-Tests	Steam, 2 lb gage	2	In-Line	0.625	1.37	7	42	10.8	0.61
3	Commercial	Steam, 5–150 lb	1–6	Stag.	0.625	1.44	8	50	10.7	0.52
4	Commercial	Steam, 5–150 lb	1–6	Stag.	0.625	1.37	7	50	10.5	0.57
5	Commercial	Steam, 5–100 lb	1–6	...	0.625	...	7	..	10.5	0.60
6	Research-Tests	Steam, 5 lb gage	4	Stag.	0.625	1.47	7	53	10.5	0.65
7	Commercial	Steam, 5–150 lb	1	Cast Iron Sections		...	...	44	10.2	0.62
8	Research-Tests	Steam, 2 lb gage	2	In-Line	0.625	1.12	6	37	10.1	0.55
9	Research-Tests	Steam, 5 lb gage	1	...	0.625	1.47	7	52	9.3	0.52
10	Research-Tests	Steam, 5 lb gage	1	...	0.625	1.37	7	56	9.1	0.49
11	Commercial	Steam, 5–150 lb	6	Cast Iron Sections		...	...	44	9.1	0.62
12	Research-Tests	Steam, 2 lb gage	1	...	0.625	1.37	7	49	9.0	0.48
13	Research-Tests	Steam, 2 lb gage	1	...	0.625	1.12	6	37	9.0	0.46
14	Commercial	Steam, 5 lb gage	1	Cast Iron Sections		...	...	...	7.0	0.50
15	Commercial	Steam, 5 lb gage	6	Cast Iron Sections		...	...	...	6.5	0.60
Air Cooling Without Dehumidification										
16	Commercial	Water, 2 fps	1–6	Stag.	0.625	1.37	7	50	8.9	0.47
17	Commercial	Water, 2 fps	1–6	Stag.	0.625	1.44	8	50	8.6	0.45
18	Reported[b]	Water, 2 fps	...	...	0.625	1.44	8	50	8.5	...
19	Research-Tests	Water, 2 fps	2	Stag.	0.75	1.75	7	55	7.2	0.45
20	Research-Tests	Water, 2 fps	6	Stag.	0.75	1.75	7	55	7.1	0.64
21	Commercial	Direct Expansion	1	...	0.75	1.62	4–6	..	5.8	0.62
22	Commercial	Direct Expansion	6	...	0.75	1.62	4–6	..	4.7	0.61
23	Research-Tests	Direct Expansion	4	Stag.	0.75	1.75	7	55	6.2	0.51
Air Cooling and Dehumidification (Approximate Only)										
24	Research-Tests	Water, 2 fps	4	Stag.	0.75	1.75	7	55	14.8[c]	0.81
25	Commercial	Direct Expansion	4	Stag.	0.625	1.44	8	50	16.0[c]	0.51

[a] Tests on an Industrial Heater, Fellows and Stewart, *Heating and Ventilating*, July, 1935, p. 36. The very high coefficient in this case is probably due to the extreme turbulence caused by the blow-through fan.
[b] Computed from data given in Rational Development and Rating of Extended Air Cooling Surface, H. B. Pownall, *Refrigerating Engineering*, Oct., 1935, p. 211.
[c] Based on wet-bulb mean temperature differences, see Table 5.

of temperature difference between the mean temperature of the air and that of the heating or cooling fluid. Table 1 gives values for the overall coefficient of several extended surface coils at 500 fpm face velocity. The items marked *research-tests* in Table 1 are the results of tests conducted by the author and his associates[4] at Case School of Applied Science, as a part of this research

[4] Prof. F. H. Vose, Head of the Mechanical Engineering Department, has encouraged this co-operative research and made the facilities of the Department available for carrying it on. Professors A. O. Willey and C. A. McKeeman have given generously of their time and effort to the work. Much credit is due to V. T. Kartorie, Assistant in Mechanical Engineering 1933-35, who submitted a thesis for a Master's Degree on the basis of his experimental work on this project. Messrs. D. E. Wise, W. B. Rust and P. Borkat have each had an important share in this research over the past two years, and have submitted undergraduate theses based on their work.

program. One of the test arrangements used in this work is shown in Fig. 1. The items marked *reported* are overall coefficients by test as reported by other investigators (see references), and the items marked *commercial* are taken from manufacturers' bulletins. Knowing the performance at one air velocity, the coefficient at any other velocity may be obtained by drawing a straight line through the given point on a logarithmic plot of overall coefficient against air velocity. Table 1 also gives the slope of such a line for each case, *i.e.*, the value of n in the equation:

$$U = C V^n, \text{ or } U_1/U_2 = (V_1/V_2)^n$$

The value of the slope n may readily be calculated from any performance table, even when the coil dimensions are not given and the overall coefficient cannot be calculated. Table 2 gives additional values of n for various types of forced-convection units.

TABLE 2. ADDITIONAL DATA ON EFFECT OF AIR VELOCITY ON HEAT TRANSFER COEFFICIENT

(As Stated in Commercial Bulletins)

TYPE No.	WORKING FLUID	No. OF ROWS	TYPE OF UNIT	n = SLOPE OF CAPACITY-VELOCITY CURVE (LOG PLOT)
26	Steam, 2 lb gage	...	Fin-Tube Unit Heater	0.66
27	Steam, 5 lb gage	...	Fin-Tube Unit Heater	0.68
28	Steam, Any Pressure	1–6	Fin-Tube Blast Coils, 7–8 Fins/in.	0.57
29	Steam, Any Pressure	1–6	Fin-Tube Blast Coils, 7–8 Fins/in.	0.55
30	Steam, 5 lb gage	1–6	Blast Coils, Continuous Fin	0.53
31	Steam, 2 lb gage	6	Blast Coils, Continuous Fin	0.51
31	Steam, 2 lb gage	1	Blast Coils, Continuous Fin	0.46
32	Steam, 5 lb gage	5	Blast Coils, C.I. with Fins	0.60
32	Steam, 5 lb gage	1	Blast Coils, C.I. with Fins	0.50
33	Steam, 5 lb gage	1	Blast Coils, C.I. no Fins	0.62
33	Steam, 5 lb gage	6	Blast Coils, C.I. no Fins	0.58
34	Dir. Exp. Refrig.	1–6	Cooling Coils, Continuous Fin	0.25
34	Water, 0.75 fps	1–6	Cooling Coils, Continuous Fin	0.24
35	Water or Brine, 1 fps	Any	Cooling Coils, Continuous Fin	0.51
35	Dir. Exp. Refrig.	Any	Cooling Coils, Continuous Fin	0.51
36	Water, 1–4 fps	Any	Cooling Coils, Continuous Fin	0.29

Tables 1 and 2 summarize most of the specific data on overall heat transfer coefficients of fin-tube units which are at present available to American engineers.

Table 3 presents a list of the methods by which the overall coefficient may be increased.

Table 4 presents a list of the methods by which the slope of the capacity-velocity curve may be increased.

It should be noted that many of the factors affecting the performance of fin-tube sections as listed in Tables 3 and 4 are dependent on natural physical laws only, and are independent of the design or proportions used by the manufacturer. (These factors were discussed at some length in the June, 1934,

paper on this research, see footnote [2].) The effect of such factors on coil performance should be determined by research and be made common property. After the laws have been determined, they should be applied in a uniform manner according to a Test Code, so that the performance of surface coils can

TABLE 3. METHODS FOR INCREASING THE OVERALL COEFFICIENT U FOR A GIVEN FIN-TUBE UNIT IN A CONSTANT-VELOCITY AIR STREAM

TYPE OF SERVICE	FLUID IN TUBES	METHODS FOR INCREASING U AT CONSTANT AIR VELOCITY
Any	Any	Placing an eddy- or turbulence-grid in front of the first row of tubes.
Any	Any	Using staggered tube arrangement (in place of in-line arrangement) to produce air turbulence.
Any	Any	Corrugating the fins or setting the fins in successive rows at different angles to produce turbulence.
Any	Any	Reducing free-open area by increasing the number of fins or the number of tubes or both.
Any	Any	Reducing the diameter of the tubes or the overall diameter of round fins (same surface area).
Any	Any	Increasing the fin thickness above the minimum now commonly used.
Any	Hot or Cold Water or Brine	Increasing the velocity of the liquid in the tubes.
Dehumidification	Any	Lowering the coil temperature or raising the dewpoint of the entering air.

TABLE 4. METHODS FOR INCREASING THE VELOCITY EFFECT OR THE SLOPE n OF THE CAPACITY-VELOCITY CURVE

TYPE OF SERVICE	FLUID IN TUBES	METHODS FOR INCREASING n, THE SLOPE OF THE CAPACITY-VELOCITY CURVE
Any	Any	Placing an eddy- or turbulence-grid in front of the first row of tubes.
Any	Any	Using staggered tube arrangement (in place of in-line arrangement) to produce air turbulence.
Any	Any	Corrugating the fins or setting the fins in successive rows at different angles to produce turbulence.
Any	Any	Increasing the number of rows or banks of tubes.
Any	Any	Reducing the ratio of fin surface to prime tube surface.
Any	Any	Increasing the fin thickness above the minimum now commonly used.
Any	Hot or Cold Water or Brine	Increasing the velocity of the liquid in the tubes.

be more accurately specified and understood by engineers generally. The results of this research project to date have demonstrated that if there is a good bond between fin and tube, the rather narrow differences in design proportions at present used by manufacturers of circular fin-tubes do not make a

great deal of difference in heat transfer coefficients compared with the differences which are due to operating conditions.

Table 5 gives recommended typical overall heat transfer coefficients for finned sections operating under various common conditions, and these values are shown graphically on rectangular coordinates in Fig. 2.

QUESTIONS TO BE ANSWERED BY RESEARCH

Assume that a typical section of a certain design has been tested according to a standard method (say with low pressure steam), and the values of U_{500} and n have been determined as already suggested. It would be desirable if

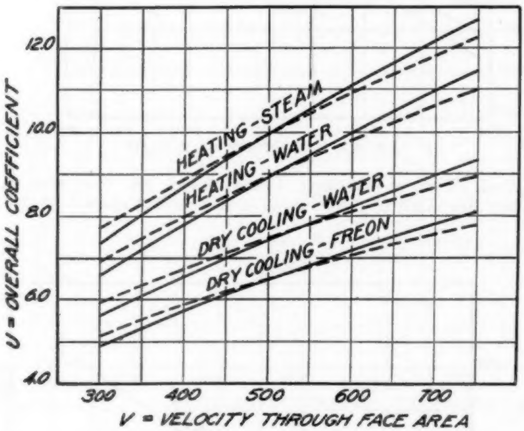

FIG. 2. GRAPHICAL REPRESENTATION OF RECOMMENDED VALUES FOR DRY-COIL COEFFICIENTS. USE DOTTED LINES FOR ONE- AND TWO-ROW SECTIONS. USE SOLID LINES FOR FOUR OR MORE ROWS.

tables could then be built to give the performance of a similar section with any number of rows of tubes, or operating under any typical condition of heating, cooling or dehumidifying. To so convert the performance under one set of conditions to that under another set of conditions will require that answers to the following questions be known:

1. What is the effect of increasing or decreasing the number of tube rows, *i.e.*, changing the depth of the coil?

2. What is the effect on the capacity of a coil if staggered tube arrangement is used, as compared with in-line arrangement?

3. For round-fin tubes, what is the effect of changing the center distance of the tubes in one row, or of changing the distance between rows?

4. What is the effect of changing the coil service from heating to cooling or vice versa?

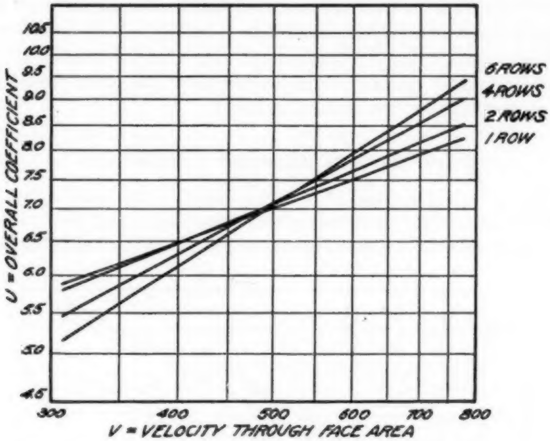

FIG. 3. TYPICAL OVERALL COEFFICIENTS FOR ONE-ROW TO SIX-ROW DEPTH SECTION (LOGARITHMIC COORDINATES).

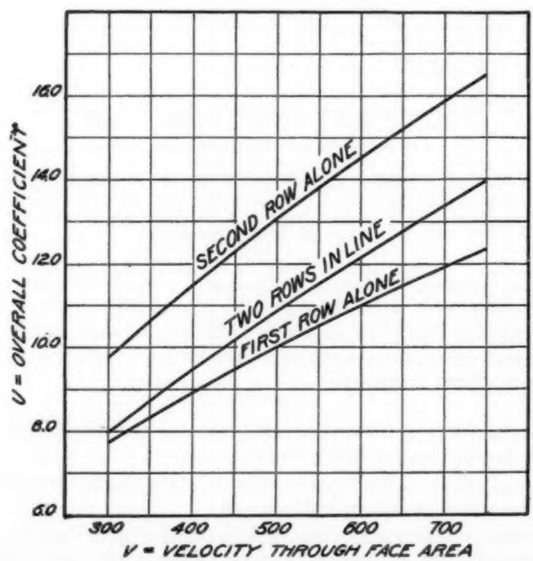

FIG. 4. CURVES SHOWING LOW EFFECTIVENESS OF FIRST OR UP-STREAM ROW OF FIN-TUBING AS COMPARED WITH SECOND ROW.

5. When a liquid is used in the tubes (water or brine), how does changing the liquid velocity change the overall coil performance?

6. How is coil performance affected by the nature of the fluid used in the tubes, *i.e.*, steam, hot water, cold water, dichlorodifluoromethane, etc.?

7. How can the dehumidifying performance at various humidities be predicted from the dry-coil performance?

The answers to these questions depend on physical phenomena, and have very little connection with the particular design used by one manufacturer or another.

These questions will now be discussed in turn, and data presented which has

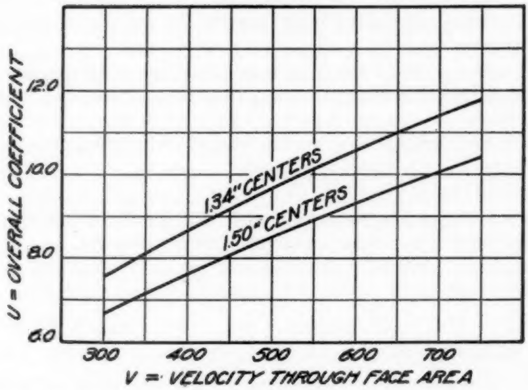

FIG. 5. RESULTS OF TESTS ON SINGLE-ROW SECTIONS WITH DIFFERENT SPACINGS OF FIN-TUBES.

been obtained in this research. An effort will also be made to correlate data from other sources which bear on the answers to the same questions.

Questions 1 and 2. Staggered Arrangement and Depth of Coil.

Increasing the number of tube-rows and staggering the tubes are two of the several methods for increasing the local air turbulence or eddying to which the coil surfaces are exposed (see Tables 3 and 4). Several investigations have demonstrated that the effect of increased turbulence is to increase both the overall coefficient U at a given air velocity and the slope n of the capacity-velocity curve. The increase in heat transfer cannot be stated as a simple percentage, in fact the capacity-velocity curves may cross as in Fig. 3, which is a typical group of curves plotted from tests on 1-row to 6-row depth of section. It will be noted that the 6-row section is the most effective when the air velocity is high, but that the 2-row section gives a slightly higher coefficient when the velocity is very low. This relation has been confirmed by tests on three types of units, and for both heating and cooling. Certain manufacturers at present use the same coefficient for any number of rows of tubes (see Units 3, 4, 16, and

17, Table 1), and several manufacturers use the same value of the slope, irrespective of the depth of section. These practices are contrary to the results of repeated tests in this research.

The effect of staggering fin-tubes is probably less than that for plain tubes, but most of the published data and photographs on the advantages of staggered tube arrangement are from the results of German investigations on banks of plain tubes. Pownall [5] gives data indicating an increase of about 11 per cent

TABLE 5. TYPICAL VALUES OF U_{500} AND n RECOMMENDED FOR USE IN ESTIMATING OR CHECKING

The following values apply to typical American fin-tubes having 4 to 8 fins per inch, on ½ in. to 1 in. tubing, with metal bond between tube and fin and for face velocities from 200 to 1000 fpm.

U_{500} = overall coefficient, Btu per sq ft per hour per degree, at 500 fpm face or approach velocity, and based on logarithmic mean temperature difference between air and fluid in tubes.

n = slope of the straight line through U_{500} when overall coefficient is plotted against face velocity on logarithmic coordinates.

= value of n in the equation: $U_1/U_2 = (V_1/V_2)^n$

It is usual practice to assume the same overall coefficient for any temperature of the entering air (heating or dry cooling), and for any coil temperature.

U_{500}		n		SERVICE	FLUID IN TUBES
1 OR 2 ROWS	4 OR MORE ROWS	1 OR 2 ROWS	4 OR MORE ROWS		
10.0	10.0	0.50	0.60	Heating	Steam, 1 to 100 lb gage
9.0	9.0	0.50	0.60	Heating	Hot water, 140 deg to 250 deg, 2 to 4 fps
7.5	7.5	0.45	0.55	Dry Cooling	Cold water, 35 deg to 70 deg, 2 to 4 fps
6.5	6.5	0.45	0.55	Dry Cooling	Direct expansion dichlorodifluoromethane. (Approximate for other refrigerants)
15.0[a]	15.0[a]	0.7	0.7	Dehumidifying	Cold water, 35 deg to 55 deg, 2 to 4 fps
13.0[a]	13.0[a]	0.7	0.7	Dehumidifying	Direct expansion dichlorodifluoromethane.

[a] Values for dehumidifying are based on wet-bulb mean temperature differences, *not* dry-bulb. This method should be used with caution, as it does not show the dry-bulb temperature changes.

in overall coefficient at approximately 400 fpm face velocity, but the number of tube rows is not stated. Results obtained by the author on a 2-row unit with close tube centers, showed only 5 per cent more heat transfer when the tubes were staggered, while a disturber-grid placed in front of the section increased its capacity 16 per cent. The increase in capacity due to staggered tube arrangement, compared with in-line arrangement, is, of course, greater as the

[5] Rational Development and Rating of Extended Air Cooling Surface. H. B. Pownall, *Refrigerating Engineering*, October, 1935, pp. 211-218.

depth of the coil is increased, and it is more important with high velocities and wide tube spacings, probably reaching about 15 per cent in some cases.

The most important loss in heat transfer due to lack of air turbulence is in the first or upstream row of tubes which receives the undisturbed air stream, and this condition is, of course, not improved by staggering the tubes. Fig. 4 gives the results of tests on a 2-row unit in which each row was tested separately and then the two rows tested together, with tubes in line. In this case the heat transfer from the second row alone was more than one-third greater than that from the first row. It has been demonstrated repeatedly in this research that the effectiveness of the first row may be brought up to that of the others by the use of a disturber-grid placed in front of the section (see data in June, 1934, paper).

Question 3. Relation of Free Area to Gross Face Area

In general, the smaller the free area for a given face area, the higher will be the heat transfer coefficient. This is because the heat transfer is dependent upon the velocity *at the surface,* and the latter is comparable to the velocity through the free-open area. There are, of course, practical limits to the reduction of free area, mainly those of air resistance and air noise, and hence the range of *percentage free area* used in practice is rather small (see Table 1).

Fig. 4 shows the results of two tests on identical 1⅜ in. o. d. fin-tubing spaced on 1.50 in. centers in one case and on 1.33 in. centers in the other. This is only a small difference in spacing, but it demonstrates that a coil is more effective if the fin-tubes are closely spaced. This is true even when comparing two points representing the same velocity through the free area. In general the best spacing is probably that which places the fins as close together as is possible without interlocking.

Question 4. Heating and Cooling Service from the Same Coil

The difficulty of comparing the heat transfer coefficients for heating and cooling is that several other conditions are also changed when the direction of heat flow is reversed. Cooling is carried out at a lower temperature level and with a smaller temperature difference than heating. Cooling fluids usually show a lower surface coefficient (inside the tubes) than is obtained with steam or hot water. These differences all tend toward a lower overall coefficient for cooling coils (see Tables 1 and 5). There are some inconsistencies in commercial practice, regarding capacities of the same coil for heating and cooling, as is shown by the fact that one manufacturer uses the same overall coefficients for heating with steam and for cooling with water (high water velocity), while another shows a 30 per cent difference between the two.

Experiments are now in progress as a part of this research, in which, for instance, 80 deg air is first cooled with 40 deg water and then heated with 120 deg water, all other conditions remaining constant.

Question 5. Effect of Liquid Velocity in Tubes

Low liquid velocity means a low coefficient U and a lower value of the capacity-velocity slope n. This is because of the high resistance to heat flow offered by the thicker viscous film of liquid at the inside tube wall when the average velocity of the liquid is reduced. The magnitude of this effect depends on the viscosity of the liquid and on the tube diameter.

There is both a rapid decrease in heat transfer and a rapid increase in friction coefficient when the Reynolds number of the liquid flow is so low that the flow changes from turbulent to viscous in character. Hence it would seem to be poor practice to operate coils with viscous flow in the tubes. To avoid the viscous region, liquid velocities in cooling coils should be well above 1 fps for water and above 2 fps for brine. For smaller tubes than ⅝ in. even higher velocities are necessary. For heating coils, on the other hand, the low limit of velocities usually depends on the allowable temperature drop of the water rather than on the encountering of viscous flow. This is because the viscosity of 160 deg water is only one-fourth as great as that of 40 deg water.

For strict accuracy the corrections for water velocity should be based on test

TABLE 6. COMPARISON OF WATER-VELOCITY CORRECTION FACTORS FOR COOLING COILS

	Value of U_{500}	Correction U_{500}	Value of n	Correction for Factor n
1. Calculated example				
Water vel. 2 fps	8.67	1.00	0.50	1.00
Water vel. 4 fps	9.66	1.11	0.55	1.11
Water vel. 6 fps	10.10	1.16	0.57	1.14
2. Commercial data No. 1				
Water vel. 2 fps	8.90	1.00	0.47	1.00
Water vel. 4 fps	9.75	1.10	0.50	1.06
Water vel. 6 fps	10.35	1.16	0.53	1.13
3. Commercial data No. 2				
Water vel. 2 fps	...	1.00	0.51	1.00
Water vel. 4 fps	...	1.01	0.51	1.00
Water vel. 6 fps	...	1.01	0.51	1.00
4. Commercial data No. 3				
Water vel. 2 fps	...	1.00	0.24	1.00
Water vel. 4 fps	...	1.04	0.24	1.00

data which give both liquid-side and air-side coefficients. To obtain such data requires a large amount of testing, however, and the corrections may be made by the simple method of changing U_{500} and n. The following example shows a method of computing the corrections. The calculated correction factors are given in Table 6. The equation for U in terms of the individual liquid-side coefficient h_1 and the air-side coefficient h_a is

$$U = \frac{1}{\frac{R}{h_1} + \frac{1}{xh_a}}$$

where x is a factor to account for temperature gradient in the fins as suggested by Pownall[6] and R is the ratio of air-side surface area to liquid-side surface[7] area.

[6] Loc. Cit. See Note 5.
[7] The above equation is Eq. 2, in the 1934 report, see footnote [1].

Typical values for substitution in this equation are: $R = 10, h_1 = 210\ V_1^{0.80}$, where V is the average liquid velocity in feet per second, and $xh_a = 0.202\ V_a^{0.65}$, where h_a is air velocity in feet per minute through the face area. The resulting correction factors based on a water velocity of 2 fps as unity, are shown in Table 6. For comparison the values given by three manufacturers of fin-tubing are included in the same table. Certain differences between the calculated correction factors and those based on commercial data are at once apparent. One error in the computation method is the assumption that x, the correction for fin temperature gradient, varies with air velocity and does not change with water velocity, but this error should be very small. Tests on this

FIG. 6. WORKING SECTION OF TEST DUCT WITH FOUR ROW DICHLORODIFLUOROMETHANE COIL UNDER TEST.

phase of the subject are as yet incomplete, but Table 6 shows the need for additional research data, and the tests are being continued.

Question 6. Effect of Changing the Fluid in the Tubes

More data are needed on surface coefficients for the fluids inside the tubes. Work is now in progress in this research on direct-expansion of dichlorodifluoromethane (see Fig. 6). The ordinary range of inside surface coefficients for steam, direct expansion refrigerants and water (above critical velocity) is probably from 250 to 1200. The coefficient should be higher for steam, but difficulties in securing adequate venting of air and uniform distribution to all tubes are often great. Pownall suggests a value of 300 for dichlorodifluoromethane and gives a range of 350 for cold water at 2 fps to 1200 at 6 fps.

The effect of a change from 250 to 1200 is to change the overall coefficient from 7.1 to 9.2 in a typical case in which the surface area-ratio is 10 and the air-side coefficient (including temperature gradient factor) is assumed constant at a value of 10, at a face velocity of 500 fpm. Thus the overall coefficient is changed only 30 per cent when the inside surface coefficient is increased almost 500 per cent. If a reasonably accurate set of inside surface coefficients were

available, an accurate set of correction factors could be set up so that a coil could be tested with steam and its capacity with all the other heating and cooling fluids calculated therefrom.

Question 7. Dehumidifying Coils

When a coil is used for dehumidifying, its capacity, based on dry-bulb temperature, is very much increased as compared with dry-cooling only, but it is difficult to show this increase by any simple method.

It will be noted from the discussion of dry-coil performance that there are at least 10 variables affecting the value of the overall coefficient, but that a few straight-line graphs against one variable, the air velocity, will serve to represent most of the common cases. The presence of dehumidification not merely adds another variable, but it adds an entire heat transfer process, *viz.*, the condensation of a vapor. This process in turn is sensitive to most of the variable conditions which affect dry cooling. In addition, it is one of the most complex of the heat transfer processes, as is shown by the current arguments regarding drop-condensation and film-condensation. Since surface heat transfer coefficients for condensation are very much greater than gas-to-surface coefficients, it is to be expected that the overall coefficient would be much increased when condensation is present, even though the two processes interfere with each other.

Methods for obtaining the performance of a dehumidifying coil have been given by Scanlan, Ruppricht, Schmidt, Knaus, King, Pownall and others,[8] and only the briefest summary can be given here. Four general viewpoints may be mentioned:

1. Since a theoretical relation exists between the surface coefficient for dry cooling and the combined coefficient for cooling and dehumidifying, the latter may be calculated from the former by means of this theoretical formula, modified by an empirical correction factor. This equation and correction factor may, if desired, be embodied in a graphical solution on the psychrometric chart.

2. The combined coefficient for cooling and dehumidification is a function of the ratio of dehumidification load to total load, and may be calculated from the dry-coil coefficient as soon as this ratio is known.

3. Dehumidification may be considered as a process separate from sensible cooling, and the coil capacity for the two calculated independently, using two separate sets of coefficients or one coefficient and a correction factor. While sensible heat removal depends on the dry-bulb mean temperature difference, latent heat removal depends on the dew-point mean temperature difference.

4. Since the wet-bulb temperature is the index of the total heat of air, the reduction of this total heat may be calculated on the basis of wet-bulb mean temperature differences and the overall coefficients based thereon, without regard for dry-bulb or dew-point temperatures.

At present there is apparently not enough research data available to establish full confidence in any one of these methods.

About 50 two-hour dehumidifying tests have been made as a part of this research, and the results have been examined according to each of the methods stated above. The conclusion is that the data are incomplete as yet, but indica-

[8] See references in Rational Development and Rating of Extended Air Cooling Surface, H. B. Pownall, *Refrigerating Engineering*, October, 1935.

tions are that the simple point-slope method used herein for expressing dry-coil performance may also be used for dehumidifying coils (method No. 4, previously mentioned), if its limitations are recognized. There has been included in Table 1 the most readily available data on U_{500} and n from these research tests and from commercial sources. U in this case is the combined overall coefficient for both sensible and latent heat, and it is based on wet-bulb logarithmic mean temperature differences. An important defect of this method is that it does not separate the latent and sensible heat loads, and absurd results may be obtained if the method is not properly used.

A satisfactory treatment of dehumidification would require an entire paper, but much additional experimental work should first be done and the results made available in a form which will permit general conclusions to be drawn.

CONCLUSIONS

Tests on fin-tube coils indicate that reasonably accurate performance tables may be built on the basis of a few simple straight-line graphs of overall heat transfer coefficient plotted against air velocity. These graphs may readily be plotted (on logarithmic coordinates) by locating the value of the coefficient at 500 fpm air velocity, and then drawing a line of given slope, as shown by Tables 1 and 5. This same method provides a means for obtaining the capacity of a unit at any operating condition when the capacity at another operating condition is known. For rapid approximate calculations, the capacity of a heating or a dry-cooling coil may be assumed to vary directly as the square root of the air velocity.

Any arrangement which increases the local eddies in the air stream will increase heat transfer, especially at the high velocities. For maximum heat transfer the fin tubes should be placed as close together as is possible without interlocking, arranged in staggered rows, and provided with a turbulence-grid in front of the section to increase the effectiveness of the first row of tubes.

Certain basic data on the performance of fin-tube coils are still incomplete. Accurate knowledge of surface coefficients for the fluids inside the tubes is lacking, and a thorough study should be made of the various methods which have been suggested for showing the performance of a coil in dehumidification.

By additional research along these lines it should be possible to develop in the near future a test code and a set of rules by which the performance of any type of heat transfer surface may be calculated, for any case of heating, cooling or dehumidifying, from a few tests made under specified simple conditions. Such a procedure would place the selection of extended surface coils upon a uniform basis and give engineers more confidence in being able to produce the results called for in their specifications.

DISCUSSION

R. H. NORRIS [9] (WRITTEN): This paper is a valuable summary of many important factors in the performance of finned-tube coils.

Fig. 3, showing the effect of varying the number of tubes deep, is a particularly useful contribution. It is rather remarkable and convenient that, although the heat transfer coefficient varies, in general, according to the depth of the coil, it appears

[9] Engineering General Department, General Electric Co., Schenectady, N. Y.

to be practically the same for all depths when the face velocity is the chosen standard of 500 fpm. It would be desirable to determine whether this independence of depth at 500 fpm may be relied on for all designs as well as for the three particular ones tested. Furthermore, it is important to know how closely the effect of depth indicated by Fig. 3 is representative of all conventional designs of finned tubing.

The use of the overall coefficient of heat transfer, U, for comparative purposes, instead of the air-to-tube coefficient, xh_a, is convenient for practical purposes when the conditions on the inside of the tube are uniform and definitely known. It is also true that determination of the air-to-tube coefficient for a single particular operating condition would complicate the test procedure. However, when the object of tests is to obtain data from which the coil performance may be predicted under all conditions of service, the use of the air-to-tube coefficient, xh_a, seems to offer considerable advantage. Suppose that, for a given coil under given air conditions, the value of xh_a is once determined, by test, as a function of air velocity, in the form: $xh_a = cV^n$. Then it is likely that the test values of c and n would be constant and applicable to all conditions of heating and cooling with simple corrections of c for the effect of air temperature. These corrections may be derived from the fact that xh_a should theoretically be a function not of velocity alone but of Reynolds' number to the nth power, $(\rho VD/\mu)^n$, where D is a characteristic dimension and ρ and μ, the air density and viscosity, are functions of the air temperature. Even where dehumidification accompanies cooling it is probable that the value of xh_a is directly applicable to the sensible portion of the heat transfer. Moreover, the use of xh_a instead of U for comparison of two different coils under similar air conditions eliminates the uncertainty or complications introduced by possible differences in the temperature and velocity of the cooling medium.

The determination of xh_a should not be unduly difficult. For a given air flow, it would be sufficient to measure U with three or four different water velocities, V_w, inside the tube. Then xh_a can be determined by plotting $1/U$ versus $\frac{1}{V_w^{0.8}}$ as explained in McAdams, Heat Transmission, page 266. Tests of this type seem preferable, for determination of xh_a, to tests with steam in the tubes. Although the steam coefficient is high, its value may be very uncertain as a result of ignorance regarding the amount of air in the steam and the uniformity of distribution of the steam.

It is to be hoped that the further research along the lines recommended by Professor Tuve will be carried out in the near future. It would be desirable if this research could include consideration of the effect of the number of rows deep on the air pressure drop as well as on the heat transfer coefficient.

F. G. HECHLER [10] AND F. C. STEWART [11] (WRITTEN): Professor Tuve is to be complimented for his contributions to the experimental data relating to heat transfer in unit heaters and coolers and also for his correlations of existing data in this field.

The data in Table 1 of the present paper show satisfactory agreement of the overall coefficient of heat transfer for fin-tube units when the fluid in the tube is condensing steam or water at an average velocity of 2 ft per second and a given temperature. For other water velocities and temperatures, however, a correction factor is required to take care of the change in the water-side coefficient. The data for cooling air when *direct expansion* is used are not so satisfactory and the values are very low. It is not always appreciated that the design of the evaporator and the method of operation as well as the refrigerant itself have a great influence on the value of the refrigerant-side coefficient. A study of the film coefficients of boiling refrigerants is being made at the Pennsylvania State College and the results now available are of interest in this connection. The data show that the coefficient is

[10] Pennsylvania State College, State College, Pa.
[11] Pennsylvania State College, State College, Pa.

higher for *dry* operation than for *flooded.* For *dry* operation the coefficient appears to remain nearly constant but for *flooded* operation it varies over a wide range depending, in the case of a vertical evaporator pipe, on the point where boiling starts as determined by an ebullator, or some other contributing cause, as well as on the tem-

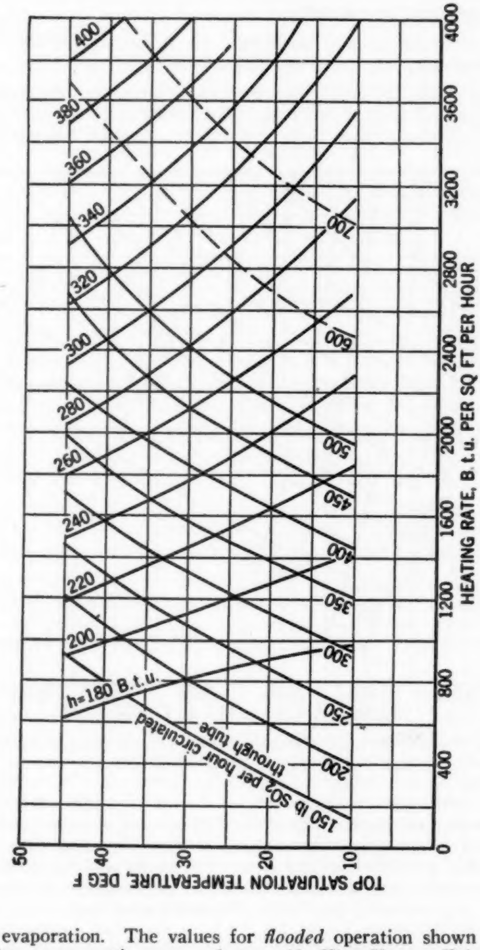

FIG. A. FILM COEFFICIENT AS A FUNCTION OF THE SATURATION TEMPERATURE FOR VARIOUS HEATING RATES

perature of evaporation. The values for *flooded* operation shown in Figs. A and B are taken from a recently prepared paper.[12] The film coefficient for the boiling refrigerant (SO_2) varies from 180 to 400 for flooded operation; for dry operation recent tests indicate that the value is about 500.

[12] Film Heat Transfer Coefficients for Sulphur Dioxide in a Vertical Evaporator, by F. C. Stewart and F. G. Hechler, *Refrigerating Engineering,* Feb., 1936.

Tests 21, 22, and 23, Table 1, show the very low values for U of from 4.7 to 6.2. Using the author's method of analysis for an area ratio $R = 10$, and an air velocity of 500 fpm, the corresponding value of h_i, the refrigerant-side film coefficient, varies from about 80 to 135 Btu per square foot per hour, indicating that this unit was not operating under favorable conditions. If by a change in operation a value of $h_i = 400$ is obtained the value of U becomes approximately 9. This represents an average increase in U of well over 50 per cent.

The tests at Penn State on film coefficients for boiling refrigerants are not yet complete and other factors are to be investigated.

In conclusion it should be noted that the results given by Professor Tuve for cooling with a boiling refrigerant emphasize the limitations of the over-all coefficient method however convenient it may be in some cases. A more rational method is to

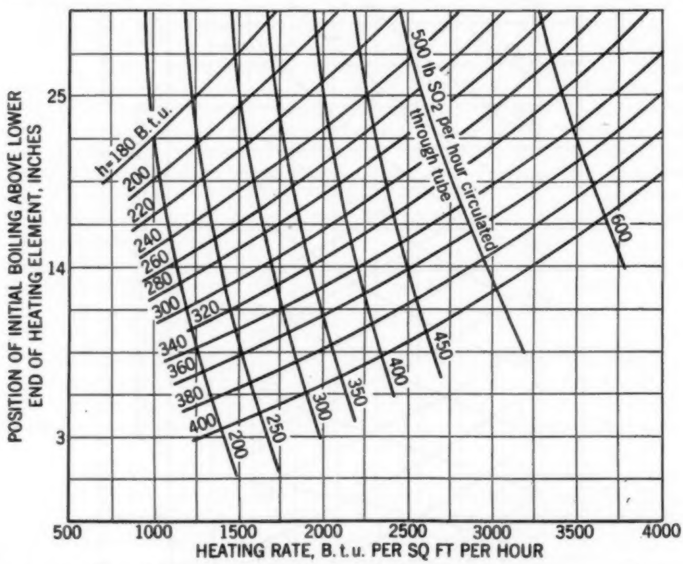

Fig. B. Film Coefficient as a Function of the Initial Boiling Position for Various Heating Rates

determine the film coefficients on both sides of the heat transfer surface for various designs, arrangements and operating conditions; these can then be combined to give the proper value of U for any designated conditions. No other procedure will so quickly and so surely give the required data. For water and for condensing steam inside pipe the film-coefficients are well determined and experiments now under way should be of help in determining the film coefficients for air and for boiling refrigerants. The effort to determine values for U by experiment for the many possible designs, arrangements and fluids would require many more tests than the film-coefficient method and can never equal the latter in adaptability.

L. O. Monroe: I would like to ask whether any test has been made to show the effect of the ratio of face velocity to the velocity through the free area or surface air velocity. You don't give any indication of the surface ratio.

F. B. ROWLEY: I would like to make a few remarks as chairman of the Research Technical Advisory Committee on Heat Transfer of Finned Tubes with Forced Air Circulation. It is always difficult in a problem of this nature to establish fundamental data without spending a lot of the Society's money for research. In this instance I would like to have the Society members know that Professor Tuve, a member of the Committee, has taken charge of the necessary research work and that it has been carried on at Case School of Applied Science without any expense whatsoever to the Society.

The author has pointed out the fact that turbulence gives an increase in the over-all coefficient of heat transfer. In Fig. 4 of the paper the effect of this turbulence in increasing transfer from the second row of tubes is brought out very clearly. I am wondering whether or not the author has any data which show the relation between pressure drop through the coil and overall heat transfer coefficient. The increase in coefficient for turbulent air flow has a definite relation to the scrubbing action of the air on the surface of the metal, which in turn increases the friction and therefore the pressure drops through the coils. There is undoubtedly a definite relation which should be of value in correlating the coefficients obtained.

I want to express the appreciation of the Committee to Professor Tuve and Case School of Applied Science for the excellent work that they have done for the Society and the hope of the Committee that the work may be continued until the fundamental coefficients are established.

J. N. HADJISKY: I should like to know if it is possible to obtain a more detailed account of the tests so that one can go over the various items. Would it be possible to publish these results in addition to the conductivity constants?

I should like to see what were the actual inlet and outlet temperatures of the cooling mediums, especially when cold water was used. Also, what were the inlet and outlet temperatures of the air?

These facts seem to me to be of more importance than the conductivity constants. In practical application, when only the conductivity constant is known, one has to assume the final temperatures, and this is not so easy or reliable.

I notice that in the test of the coils with cooling water, the tube diameters are 3/8 in., 5/8 in. and 3/4 in. For a given velocity of water in the coil, say, one foot per second, the conductivity value for the three different diameters will not be the same because of the different area and turbulence of flow. Has the author anything to say on that item?

G. L. TUVE: Mr. Norris' written discussion mentioned the effect of coil-depth on the performance. We would like to exchange information with any one else who has data along those lines. Professors Hechler and Stewart have discussed the subject of cooling accompanied by dehumidification. I forgot to mention that subject, but four methods of calculating cooling and dehumidification are briefly described in my paper. We have made some studies on direct expansion, and I am glad to hear that Pennsylvania State is making a thorough investigation of that phase of the subject. We shall exchange information with them. One of the four methods now in use for calculating the performance of a dehumidifying coil should be selected as a standard method by engineers, so we would not have so many people telling us how to do the same thing in different ways.

In answer to Mr. Monroe, we previously plotted all data on the basis he suggested, but we felt that since most practical engineers deal with duct velocity rather than velocities through the free area, we would rather present the data in that way in our paper.

We do not have complete data on air friction or resistance, and that should be a subject of further study, as Professor Rowley has suggested.

Mr. Hadjisky's remarks on the detailed calculations show that he has tried some of them, and that the manufacturers' work along this line isn't entirely in vain. After you have tried a few of those detailed calculations, you can appreciate these many-page tables published by the manufacturers. We did not have room for such tables, although we have made plenty of those calculations.

I am sorry we could not give the data in a more complete form, but we felt that most engineers prefer to make calculations on the basis of the overall coefficient U.

No. 1030

SUBJECTIVE REACTIONS OF HUMAN BEINGS TO CERTAIN OUTDOOR ATMOSPHERIC CONDITIONS

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NEW HAVEN, CONN.

SOURCES OF BASIC DATA

THE study here reported was primarily undertaken in the attempt to discover whether any subtle and hitherto unrecognized climatic factors might exert a demonstrable effect upon subjective reactions as to the pleasantness or unpleasantness of the outdoor atmosphere. Data on this point were supplemented by a study of daily variations in the incidence of illness. The basic data utilized may be summarized as follows:

1. Records taken at 8:00 a.m. of air temperature, temperature change from preceding day at the same hour, relative humidity, sunlight (expressed in percentage of full sunshine for the hour 8-9 a.m.), sunshine change (as compared with the same hour on the preceding day), wind movement, barometer, and barometric change (as compared with the same hour on the preceding day). These data, for all week days for the period December 1933 to February 1935, inclusive, were courteously furnished by Director Cornelius Doherty, of the United States Weather Bureau Station at New Haven.

2. Records of ions present in the air outside the John B. Pierce Laboratory in New Haven at 9:00 a.m. each morning from January 1934 to February 1935, inclusive. These data have been discussed from a physical standpoint by Gagge and Moriyama (1935)[1] with a description of the apparatus and methods used. For the purpose ot the present paper the values are grouped according to sign and size of ions. *Light Ions* have been classed as those collected by the counter at a threshold of 0.07 centimeter per second; volt per centimeter. This procedure collects all the ions having a mobility above the value stated, with a considerable proportion of the large ions as well. Only one-fifth to one-half of the ions measured are actually of a mobility below 0.07 centimeter per second; volt per centimeter, but the *Light Ions* group is generally comparable with the results obtained by Yaglou and his associates (Yaglou, Benjamin and Choate, 1931; Yaglou, Brandt and Benjamin, 1933),[2] but with somewhat more large ions than he obtained. The *Total Ions* include all those with a mobility greater than 0.0006 centimeters per second; volt per centimeter.

3. For the whole period of the study there was obtained from a group of volunteers in New Haven an expression of their general reaction to the outdoor weather condi-

* Director, John B. Pierce Laboratory of Hygiene.
** Asst. Director, John B. Pierce Laboratory of Hygiene.
[1] The Annual and Diurnal Variations of Ions in an Urban Community, A. P. Gagge and I. M. Moriyama, *Journal of Terrestrial Magnetism and Atmospheric Electricity*, *40*, 295, 1935.
[2] Changes in Ionic Content of Air in Occupied Rooms Ventilated by Natural and by Mechanical Methods, C. P. Yaglou, L. C. Benjamin and S. P. Choate, A. S. H. V. E. TRANSACTIONS, Vol. 38, 1932, p. 191.
Observations on a Group of Subjects Before, During and After Exposure to Ionized Air, C. P. Yaglou, A. D. Brandt, and L. C. Benjamin, *Journal of Industrial Hygiene*, *15*, 341, 1933.

Presented at the 42nd Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Chicago, Ill., January, 1936, by L. P. Herrington.

tions as those conditions affected them on their way to their offices in the morning. Each person willing to aid in this study was provided each week with a record form ruled with five spaces for each day of the week (except Sunday), headed respectively *Very Pleasant, Pleasant, Indifferent, Unpleasant,* and *Very Unpleasant*. They were asked to "indicate simply by a check your reactions to the weather in the proper column." Reports were obtained from 40-100 persons on a given day. On Saturdays the group occasionally dropped to 30, but the number of votes per day for the whole period averaged 80. It is desired to express special appreciation to the Security Insurance Co., The Southern New England Telephone Co., and the Department of Public Health of the Yale Medical School for their cooperation in securing these volunteers.

For the purpose of analysis the five possible votes have been numbered by decades from 10 to 50, 10 corresponding to *Very Unpleasant* and 50 to *Very Pleasant*. A high value, therefore, means a pleasurable reaction. It is very clear from examination of the figures that whatever this expression of sentiment meant, it did represent a subjective reaction which was quite definite and quite consistent within the group. In order to test this consistency the total group for a sample period of two months was divided into two sub-groups according to sex. The correlation between votes of the males and females for this period was 0.936 with a probable error of 0.012. Therefore, it is clear that a very real subjective phenomenon was being dealt with, influenced only to a minor degree by individual variations, and unaffected by sex.

4. Through the courtesy of the Southern New England Telephone Co. (Mr. Edward Dejon) records were furnished of daily absences, classified according to cause as reported by the individual, for the period December 1933-March 1935. The population covered included about 1400 employees of the Company in the New Haven area. Through the courtesy of Dr. Leonard Greenburg, Health Officer of the City of New Haven, records were provided of school absenteeism, similarly classified for the period December 1934-March 1935. This covered a school population of about 35,000.

The mean monthly averages of data as to meteorological conditions, ions and votes as to pleasurableness are summarized in Table 1.

FUNDAMENTAL METEOROLOGICAL DATA

As a background for the studies a word should be said as to the basic meteorological conditions which prevailed in New Haven during the period of observation. The monthly mean values in Table 1 show that the winter of 1933-34 was an unusually cold one with a mean temperature of 13 F, for the whole month of February, 1934. On two days in December, 1933,[3] and on one day in February, 1934, temperatures below zero were recorded (—8 deg, —6 deg and —14 deg respectively). During February, 1934, only six days showed temperatures over 20 deg and the maximum was 34 deg. During the three months' period (December-February) temperature changes from one day to another (in both cases at 8:00 a.m.) were often marked. In December, 1933, there was one increase of 27 deg. In January there were two increases of over 20 deg (22 deg and 23 deg), and two decreases of over 20 deg (22 deg and 32 deg). In February there were three increases of over 20 deg (24 deg, 26 deg and 30 deg) and one decrease of the same magnitude (32 deg).

Relative humidity for these three months (December, 1933-February, 1934) averaged between 67 and 75 per cent and ranged from 31 per cent to 98 per cent. During the three months there were 11 values below 50 per cent and 13 values above 90 per cent. The percentage of sunshine averaged only 22 for January, 1934. Wind movement for the three months averaged 9-11 miles per hour. On one day in January and two days in February it exceeded 20

[3] To avoid repetition it should be noted that all data here cited refer to 8-9:00 a.m., and that Sundays are omitted from consideration.

miles (21, 25 and 26 miles). Barometric changes were often considerable. One day in January there was a drop of 21 millimeters (as compared with the previous day at the same hour), and on two days in February there were increases of over 20 millimeters (22 and 25).

During the Spring the usual changes occurred, rising temperature, increasing sunshine, decrease in wind movement, but no great average change in relative humidity. Mean monthly temperature was highest in July (72 deg). The maximum for this month was 81 deg, the minimum 65 deg. Sunshine showed monthly maxima in March (50 per cent), April (57 per cent), and August (54 per cent). Wind movement for the summer months averaged 7-8 miles, with only one or two days a month over 10 miles. In July no day exceeded 11 miles. Relative humidity reached a peak in September and November (78 and 80 per cent respectively). In September only four days fell below 75 per cent.

Table 1. Mean Monthly Values for Basic Meteorological Data, Ion Counts and Votes as to Pleasurableness (8–9 a.m.)

Month	Vote	Temp. F	Rel. Humidity %	Sunshine %	Wind Miles per Hr	Bar. MM	Ions per C.C. Light Positive	Light Negative	Total Positive	Total Negative
Dec. 1933	26	26	71	27	10	762	...	...	...	...
Jan. 1934	23	27	75	22	9	763	396	224	11100	12100
Feb.	24	13	67	36	11	767	351	287	12500	13650
Mar.	21	33	71	50	10	770	287	285	11740	12180
April	21	48	73	57	9	765	303	273	8900	8930
May	22	57	70	41	7	763	391	251	6980	7530
June	22	67	71	45	8	760	516	204	4830	5040
July	22	72	73	49	7	762	542	108	6060	5080
Aug.	20	66	71	54	7	764	337	263	5720	5910
Sept.	24	59	78	23	8	760	252	143	5840	5890
Oct.	20	50	73	47	9	763	448	324	10100	12250
Nov.	25	49	80	45	10	765	366	305	11400	12290
Dec.	24	23	68	33	10	764	319	248	10880	10780
Jan. 1935	26	23	68	46	9	766	345	286	12970	13760
Feb.	23	26	72	46	9	762	307	236	11320	13550

The winter months of December, 1934-February, 1935, were much milder than the corresponding months of the previous year, particularly February.

The intercorrelations for the daily data for the period March, 1934, to February, 1935, inclusive, are summarized in Table 2. In this computation the first three months (December, 1933-February, 1934) are omitted, but separate correlations for each three-month period of the 15 months studied will be discussed later. With a basis of 300 days for computations, the probable error for various correlations is as follows:

Correlation	Probable Error
0.1–0.3	0.04
0.4–0.6	0.03
0.7	0.02
0.8–0.9	0.01

TABLE 2. SUMMARY OF INTER-CORRELATIONS BETWEEN DAILY OBSERVATIONS MARCH, 1934–FEBRUARY, 1935

WEATHER FACTORS	VOTE	TEMP.	TEMP. CHANGE	REL. HUM.	SUN-SHINE	SUN CHANGE	WIND	BAR.	BAR. CHANGE	LIGHT POSI-TIVE	LIGHT NEGA-TIVE	TOTAL POSI-TIVE
Temperature...	−.04	...	...	...	...	...	...	...	...	...	...	...
Temp. Chg....	−.18	.23	...	...	...	...	...	...	...	...	...	
Rel. Hum......	−.64	.24	.26	...	...	...	...	...	...	...	...	...
Sunshine.......	.78	−.13	−.20	−.55	...	...	...	...	...	...	...	...
Sun. Chg.......	.49	−.05	−.20	−.34	.64	...	...	...	...	...	...	...
Wind..........	−.09	−.19	−.15	−.11	−.08	.02	...	...	...	...	...	...
Barometer......	−.18	−.31	−.16	−.09	.26	.07	−.11	...	...	...	...	...
Bar. Chg.......	.41	−.18	−.55	−.44	.43	.37	.00	.34	...	...	...	...
Lt. Ions Positive	.15	.24	−.20	−.21	.18	.10	.00	−.12	.16	...	...	...
Lt. Ions Negative.........	.03	−.16	−.18	−.24	.13	.07	.08	.07	.22	.30	...	...
Total Ions Positive.........	−.35	−.34	.16	.26	−.20	−.13	−.23	.17	−.15	−.10	.16	...
Total Ions Negative....	−.34	−.36	.08	.25	−.22	−.06	−.18	.15	−.01	−.10	.19	.84

Therefore, all correlations of 0.15 or over are more than three times their probable error and may be considered statistically significant.

Ignoring for the present the votes and ions, it will be noted that temperature is negatively correlated with the barometric reading, barometric change, wind and sunshine, and is positively correlated with temperature change and relative humidity. Temperature change shows the same general relations but has a very high correlation with barometric change (—0.55). Relative humidity is positively correlated with temperature and temperature change, and negatively correlated with sunshine (—0.55), barometric change (—0.44) and sunshine change. Sunshine is positively correlated with barometer and barometric

TABLE 3. BASIC METEOROLOGICAL DATA, ION COUNTS, AND COMFORT VOTES, ACCORDING TO WIND DIRECTION. VALUES GIVEN REPRESENT MEAN EXCESS FOR GIVEN PERIOD OF LAND-WIND DAYS OVER SEA-WIND DAYS

PERIOD	VOTE	TEMP. F	REL. HU-MIDITY %	SUN-SHINE %	WIND MILES PER HOUR	BAR. MM	LIGHT POS. IONS	LIGHT NEG. IONS	TOTAL POS. IONS	TOTAL NEG. IONS
Dec. '33-Feb. '34	0	−9	− 8	16	2	7	90	100	−4000	−3200
Mar.-May, '34..	+3	−4	−10	19	1	5	16	72	590	350
June-Aug. '34..	+5	−3	−11	19	0	2	29	38	− 900	10
Sept.-Nov. '34..	+7	−5	−11	8	0	3	15	−24	−3730	−6500
Dec. '34-Feb. '35	+2	−9	−11	20	0	4	9	25	−3600	−4100

change, and negatively correlated with temperature change and relative humidity. Sunshine change is positively correlated with sunshine (0.64), and shows the same relations as sunshine to the other variables. Wind movement is negatively correlated with temperature and temperature change. Barometer and barometric change are positively correlated with sunshine and with each

other, and both are negatively correlated with temperature, temperature change and relative humidity.

These relationships are all fairly obvious, but are merely reviewed here in relation to their mutual influences on the sensations of pleasantness and the ionic content of the air.

One other important problem of a basic nature remains to be considered; that of wind direction. In New Haven, the important factor here is the neighborhood of Long Island Sound to the South. We have classed winds from the west, north-west, north and north-east as land-winds, and all others as sea-winds. The results of this analysis are presented in Table 3 by three-month periods.

It will be noted that the figures in Table 3 are stated in terms of mean excesses of values for land-wind days over sea-wind days. It is apparent that the (northerly) land-wind days are *always cooler* and dryer and more sunny than the (southerly) sea-wind days. They were slightly more windy in Spring and had higher barometric readings. The effects on temperature were most marked in winter. The land-wind days were on the whole more comfortable except in the first winter quarter.

Light ions were very slightly higher on land-wind days. Total ions were very markedly lower for the land-wind days in fall and winter.

Ionic Content of the Atmosphere

With this general meteorological background the new data obtained may be considered with respect to ionic content.

The data in Table 2 for the year as a whole bring out certain general relationships, but the problem is, of course, greatly complicated by broad seasonal influences which it is important to distinguish from effects of day-by-day variations. In Table 4 data are therefore presented correlations between ionic content and various meteorological factors for each quarter as well as for the entire year.

It should be noted that for a single quarter (of 75 days) the probable error for various correlations is:

Correlation	Probable Error
0.1–0.2	0.08
0.3–0.4	0.07
0.5	0.06
0.6	0.05
0.7	0.04
0.8	0.03
0.9	0.01

Therefore, for a single quarter, only correlations of over 0.25 are statistically significant.

In attempting to analyze the complex relationships involved, chief stress may be placed on total ions since—as pointed out above—the group of small ions as recorded in the tables includes a very considerable and highly variable proportion of large ions. A detailed study of the actual number of small and intermediate ions is being made by determining the characteristic curves for

TABLE 4. CORRELATIONS OF IONIC CONTENT WITH VARIOUS METEOROLOGICAL DATA BY SEASONS

Ion Groups	Temp.	Temp. Chg.	Rel. Hum.	Sun-shine	Sun Chg.	Wind	Bar.	Bar. Chg.	Light Ions Positive	Light Ions Negative	Total Ions +	Total Ions −
LIGHT IONS POSITIVE												
March-May	+.20	−.23	+.15	+.13	+.34	+.14	−.26	+.08	...	+.73	−.50	−.32
June-August	−.02	−.29	−.03	+.17	+.04	+.20	−.13	+.19	...	+.10	+.07	−.02
September-November	−.27	−.39	−.45	+.22	+.06	+.11	−.25	+.27	...	+.53	+.07	+.20
December-February	.00	−.44	−.03	−.02	−.03	−.19	+.05	+.08	...	+.55	+.33	+.30
Year	+.24	−.20	−.21	+.18	+.10	.00	−.12	+.16	...	+.30	−.10	−.10
LIGHT IONS NEGATIVE												
March-May	−.20	−.36	−.60	+.69	+.33	+.30	+.03	+.39	+.73	...	−.30	−.42
June-August	−.16	−.01	−.12	+.11	−.04	+.07	+.05	+.21	+.10	...	+.37	−.15
September-November	−.18	−.16	−.35	+.04	−.06	+.08	−.09	.00	+.53	...	+.34	+.27
December-February	+.69	−.37	.00	+.75	+.13	−.20	+.13	+.31	+.85	...	+.31	+.33
Year	−.16	−.18	−.24	+.13	+.07	+.08	+.07	+.22	+.30	...	+.16	+.19
TOTAL IONS POSITIVE												
March-May	−.40	+.23	+.29	−.25	−.31	+.01	+.35	−.17	−.50	−.30	...	+.79
June-August	−.03	−.13	+.35	−.24	−.22	+.21	+.20	−.02	+.07	+.37	...	+.03
September-November	−.20	+.33	+.11	−.15	−.27	−.40	+.01	−.26	+.07	+.34	...	+.86
December-February	+.24	+.17	+.40	−.19	−.06	−.57	+.16	−.17	+.33	+.31	...	+.95
Year	−.34	+.16	+.26	+.20	−.13	−.23	+.17	−.15	−.10	+.16	...	+.84
TOTAL IONS NEGATIVE												
March-May	−.41	+.24	+.25	−.65	−.19	−.13	+.24	−.21	−.32	−.42	+.79	...
June-August	−.20	−.04	+.07	−.01	−.03	+.05	+.37	+.31	−.02	−.15	−.03	...
September-November	−.25	+.27	−.01	−.21	−.29	−.29	+.02	−.16	+.20	+.27	+.86	...
December-February	+.29	+.17	+.50	−.26	−.22	−.54	+.11	−.15	+.30	+.33	+.95	...
Year	−.36	+.08	+.25	−.22	−.06	−.18	+.15	−.01	−.10	+.19	+.84	...

each day's observations (which were made at four critical thresholds). A general survey of Table 4 makes it clear, however, that in general when the total ions are positively correlated with a given meteorological condition the small ions are negatively correlated with that condition, and vice versa. This is what one would expect from the findings of Wait and Torreson (1934)[4] that the number of small ions in the atmosphere is in a large degree controlled in inverse fashion by the number of large ions present. The fact that correlations between light ions and total ions do not appear in Table 4 may be explained by the mixed nature of our small-ion group and by the general tendency of factors tending to increase all classes of ions to mask the relative relationships involved.

For immediate purposes, then, the study may be limited to a consideration of total ions and also to ions of one sign since inspection of the table shows that both positive and negative total ions follow the same course. The correlation between total positive and total negative ions is between 0.79 and 0.95 for three of the four quarters, and is 0.84 for the year as a whole. In the summer quarter alone does this correlation fail. Furthermore, if one compares the quarterly correlations of the various meteorological factors with positive and negative ions respectively for the various quarters, one observes almost identical relations.

The following relationships are strikingly apparent for both positive and negative total ions:

Total ions increase

1. With decrease in atmospheric temperature (except for winter quarter).
2. With increase in relative humidity (except for negative ions in the fall quarter).
3. With decrease in sunshine.
4. With decrease in windiness (except for the summer quarter).
5. With increase in barometric reading.

In order to determine the relative significance of these five factors, partial correlations were computed which are presented in Table 5.

It is clear that the correlation with sunshine is eliminated by holding relative humidity constant while the correlation with barometric pressure disappears when temperature is held constant. There remain three other correlations which are unaffected by holding other variables constant. It may be concluded that the total ionic content of the atmosphere is highest on cool days, on days of high humidity, and on comparatively windless days; and that each of these tendencies is independent of the other two. All these phenomena are susceptible of reasonable explanation. The higher ionic content on cool days is consistent with the higher ion counts obtained in winter, and with the normal rise observed toward evening. When the air is cool the ions present tend to accumulate near the ground and furthermore, the tendency of ions to pass from the chimneys, surfaces and openings of warm buildings into the cool air would be increased. That this phenomenon did not occur in the winter quarter (see Table 4) is surprising, but the number of observations for a single quarter necessarily yields unreliable results. The increase of ions on relatively windless days is, of course, to be expected since wind, like vertical air currents on warm days,

[4] The Large-Ion and Small-Ion Content of the Atmosphere at Washington, D. C., G. R. Wait and O. W. Torreson, *Journal of Terrestrial Magnetism and Atmospheric Electricity*, 39, 111, 1934.

would tend to disperse ions. Finally, the increase of ions on moist days suggests—as would be expected—that particles of moisture vapor constitute a part of the ion population of the air.

It must be remembered that all these observations were made in a crowded city street with ample opportunity for contamination with ions from chimneys and automobile exhaust. To determine how great this effect might be, check determinations were made on 16 days in October, November and December, 1934, on the top of West Rock, a hill 400 ft high and situated in reasonably open country three miles from the Laboratory. After the regular determination had been made at the Laboratory, the ion-counter was taken out to the hilltop and connected with the vacuum line operating the windshield wiper on an automobile, and a second determination was made one or two hours after the first. On 10 days the total ion count was higher at the Laboratory and on six days higher on West Rock. The averages for the 16 days were:

TABLE 5. CORRELATION OF TOTAL POSITIVE IONS WITH VARIOUS METEOROLOGICAL CONDITIONS WITH OTHER CONDITIONS HELD CONSTANT

CORRELATION WITH	FACTORS HELD CONSTANT					
	None	Temp.	Rel. Hum.	Wind	Sunshine	Bar. Pressure
Temperature	−.34	...	−.43	−.40	−.39	−.31
Rel. Humidity	+.26	+.37	...	+.24	+.18	+.28
Wind	−.23	−.32	−.21	...	−.26	−.22
Sunshine	−.20	−.26	−.07	−.23	...	−.26
Bar. Pressure	+.17	+.07	+.20	+.15	+.24	...

TABLE 6. COMPARISON OF ION COUNTS AT URBAN AND RURAL STATIONS

STATION	TOTAL IONS		LIGHT IONS	
	Positive	Negative	Positive	Negative
Laboratory	7900	8450	462	338
West Rock	6290	5160	448	547

The total positive ions, as will be noted, averaged about 25 per cent higher in the city and the total negative ions over 60 per cent higher than for the hilltop outside. The data obtained are therefore significant only for the city itself and not for the region as a whole. It was, of course, correlations with sensations of pleasure in the city which were of interest and such data are obviously of importance since so large a proportion of the human race does live in cities.

SUBJECTIVE SENSATIONS AS TO PLEASUREABLENESS

The analysis of results of the expressions of opinion as to the pleasantness of the outdoor weather conditions each day are of interest at this point. The

correlations of expression of opinion with various meteorological conditions for each of the five three-month periods studied and for the complete calendar year March, 1934, to February, 1935, inclusive, are presented in Table 7.

A general survey of the table brings out the following general relationships:

1. The weather is more likely to be pronounced pleasant in fall and winter with decreased temperature, but shows no such relation in spring and summer.
2. The judgment of pleasantness increases very markedly with decrease in relative humidity (—0.64 for the year).

TABLE 7. CORRELATION OF VARIOUS METEOROLOGICAL DATA WITH VOTES AS TO PLEASANTNESS OF ATMOSPHERIC CONDITIONS. BY SEASONS

Calendar Period	Temp.	Temp. Chg.	Rel. Hum.	Sunshine	Sun Chg.	Wind	Bar.	Bar. Chg.	Light Ions +	Light Ions −	Total Ions +	Total Ions −
Dec. '33-Feb. '34.	−.12	−.27	−.29	+.51	+.28	−.32	+.06	+.39	...	...	...	...
Mar.-May '34....	−.02	−.03	−.66	+.83	+.59	−.14	+.07	+.36	+.22	+.41	−.29	−.19
June-Aug. '34....	−.08	−.07	−.79	+.75	+.46	−.26	+.22	+.36	+.09	−.25	−.35	−.22
Sept.-Nov. '34...	−.36	−.41	−.66	+.78	+.48	−.08	+.31	+.58	+.19	+.11	−.27	−.19
Dec. '34-Feb. '35.	−.38	−.26	−.52	+.22	+.40	−.03	+.17	+.42	−.14	−.16	−.45	−.56
Mar. '34-Feb. '35.	−.04	−.18	−.64	+.78	+.49	−.09	+.18	+.41	+.15	+.03	−.35	−.34

3. It increases still more markedly with increase of sunshine (+0.78 for the year).
4. It increases with decrease in wind velocity in two of the five quarters.
5. It increases with rising barometer (the mean value for the year (+0.18).
6. It increases with a decrease in total ions of either sign (—0.35 for the year).

The absence of any correlation between the sensation of pleasureableness and temperature, except in the fall and winter months of 1934-35 is somewhat surprising. It is probably accounted for by the fact that votes were taken early in the morning when no extremely high temperatures occur. No doubt votes taken in the afternoon would have shown an inverse relation with temperature in summer. It is also somewhat unexpected to find that in winter cold mornings were voted more pleasant, but the correlation can presumably be explained by the bracing effect of such mornings. It seems quite evident from the picture as a whole that the human body acclimates itself (and clothes itself) in such a way that standards are rather closely adapted to the temperature to be expected at a given season. If this were not the case, there would be some correlation between the votes and temperature for the year as a whole.

A significant negative correlation with wind velocity appears in two quarters, December, 1933-February, 1934, and June-August, 1934. No obvious reason appears for this relationship and it may be due merely to chance.

It should be noted that in all the correlations so far presented, changes in temperature, sunshine and barometric pressure, have been tabulated with regard to sign—a marked rise being placed at the top of the correlation table and a marked fall from the preceding day at the bottom. To check the possibility of variability, *per se*, correlations were also computed for the mean votes as to

pleasureableness with regard to these variables, irrespective of sign, and taking into account only the actual magnitude of deviation from the preceding day. The correlations obtained were —0.00 for sunshine change, +0.08 for barometric change, and +0.03 for temperature change, none of them, of course, being statistically significant.

There remain four correlations which are consistent and reasonably high throughout the year: a positive correlation with sunshine (and incidentally with sunshine change); a negative correlation with relative humidity; a positive correlation with barometric change (and incidentally with barometric pressure); a negative correlation with total ions. Partial correlations for these four variables are presented in Table 8.

It appears evident that the correlation with barometric change is a secondary one since it disappears when sunshine is held constant. The other three relationships seem to be primary and independent.

It is clear that the dominant factor in the agreeableness of the weather is—as might be expected—sunshine. The zero-order correlation is about +0.8 and

TABLE 8. CORRELATION OF MEAN VOTES AS TO PLEASANTNESS WITH VARIOUS METEOROLOGICAL CONDITIONS WITH OTHER CONDITIONS HELD CONSTANT

CORRELATION WITH	FACTORS HELD CONSTANT				
	None	Sun-shine	Rel. Hum.	Bar. Chg.	Total + Ions
Sunshine	+.78	...	+.67	+.73	+.77
Relative Humidity	−.64	−.40	...	−.56	−.61
Barometric Change	+.41	+.01	+.19	...	+.40
Total Positive Ions	−.35	−.32	−.25	−.32	...

is reduced only to about +0.7 when relative humidity is held constant. The correlation with relative humidity is also a real one. Its zero-order value is —0.6 which is still —0.4 when sunshine is held constant. In order to make sure that this phenomenon was not conditioned by reactions to days of actual rain, correlations for the 257 days on which there was not even a trace of precipitation between 8 and 9 a.m. were re-computed. The correlations remained essentially identical with those presented in Table 8.

There remains the negative correlation with total ions which reveals an unsuspected relationship. The correlation is consistently negative, both for positive and negative ions at all seasons of the year. It is of the order of —0.3 and is unaffected by holding sunshine or relative humidity or barometric change constant. It should be emphasized that this phenomenon has nothing to do with the sign of the ions and lends no support to the theory of Dessauer that positive ions are objectionable. Indeed, since total positive and total negative ions show a mutual correlation of +0.8, no separate influence of the two groups of total ions could be expected. It seems clear, however, that days on which the total ion content of the air is high are judged to be unpleasant, irrespective of sunshine and relative humidity. Since the ions in city air are presumably largely made up of vapor particles from heating plants, automobile exhausts and the like, it is believed that the observed correlation is most probably due to the

effect of such substances in the atmosphere. If this conclusion is justified, it gives perhaps, new evidence of the subtle influence of odor upon human welfare.

To check these conclusions from another line of approach, the average characteristics of the most satisfactory and the least satisfactory days, as indicated by the votes of the subjects were computed. There were 43 days with mean votes of 48, 49, and 50. These were averaged as the best days while 41 days with votes ranging from 12 to 23 were taken for the worst days (10 represents the lowest possible vote, 50 the highest). The major differences were as given in Table 9.

TABLE 9. GROUP REACTIONS TO MOST SATISFACTORY AND LEAST SATISFACTORY DAYS

Group Reaction	Sunshine Per Cent	Per cent Days on which Rain Fell[a]	Relative Humidity	Total Ions	
				Positive	Negative
Best Days	87	0	63	7240	7740
Worst Days	0	80	92	14100	15220

[a] Between 8 and 9 a.m. Rainy days include those on which even a trace of rain fell at this hour.

The pleasant days had also a slightly higher temperature (56 deg against 50 deg), slightly less wind (8 miles per hour as against 9), and a slightly higher barometric pressure (765 as against 761). Temperature on the pleasant days varied from 35 to 76 deg and on the unpleasant days from 26 to 75 deg; wind varied on the pleasant days from 3 to 16 miles and on the unpleasant days from 3 to 21 miles per hour. The really consistent conditions were sunshine, relative humidity and ions. The pleasant days were fair (only 7 out of 43 with less than 80 per cent sunshine) and the unpleasant days were rainy. The higher ion-content of the air in this comparison may in part have been due to moisture droplets, but the results obtained by partial correlation show that high ion counts are associated with unpleasantness even when relative humidity and sunshine are held constant.

SICKNESS RECORDS

Finally, the data obtained from the Southern New England Telephone Co. and the New Haven Board of Health, in regard to illnesses causing absences from work and from school, may be considered respectively. For the telephone company records are available of absences due to respiratory diseases for the whole year from December 1933 to February 1935. For the schools similar data are available for the three months' period December 1934 to February 1935. These were classified as to the day on which absence was first recorded.

The first thing that was obvious on analysis of both sets of data, was the tremendous excess of absences recorded for Mondays. From the school data for the whole three months the mean value for Monday was 55 per cent in excess of the weekly mean. Tuesday, Wednesday, Thursday, and Friday were respectively 16 per cent, 3 per cent, 33 per cent and 9 per cent below the weekly mean. This is, of course, a result of the fact that absences recorded in the school and business office as beginning on Monday include also those illnesses

which really *began* to be incapacitating on Saturday and Sunday. Therefore by combining the mean weekly distribution with the actual mean value for each week a normal expectancy was computed and an attempt was made to analyze deviations from this mean expectancy with relation to various meteorological conditions for the quarter of the year. Correlations between the observed meteorological conditions on the day before and the third day before the beginning of absences were also determined. The general seasonal variations which are obvious and familiar were not of interest but all efforts to obtain, by

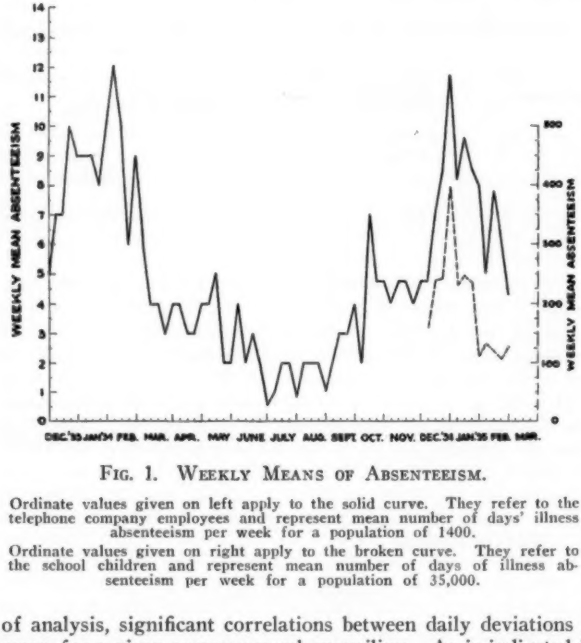

FIG. 1. WEEKLY MEANS OF ABSENTEEISM.

Ordinate values given on left apply to the solid curve. They refer to the telephone company employees and represent mean number of days' illness absenteeism per week for a population of 1400.

Ordinate values given on right apply to the broken curve. They refer to the school children and represent mean number of days of illness absenteeism per week for a population of 35,000.

methods of analysis, significant correlations between daily deviations from the expected mean for a given season proved unavailing. As is indicated in Fig. 1, which presents the weekly means for both school and telephone company, there are two major factors involved in the prevalence of respiratory diseases causing absences—the general seasonal swing and the epidemic curve, which culminates especially in certain particular weeks. These two factors, so far as our data are concerned, appear to mask very effectively any specific effects of the meteorological conditions on a given day. Gover, Reed and Collins (1934),[5] in a study of more accurate data (daily reports from groups of students in six cities) find that the incidence of respiratory illness is associated with low temperature, irrespective of season but particularly in early fall. High relative humidity, subnormal daily temperature range and low sunshine were also slightly correlated with high disease rate.

[5] Time Distribution of Common Colds and Its Relation to Corresponding Weather Conditions, M. Gover, L. J. Reed, and S. D. Collins, *Public Health Reports*, 49, 811, 1934.

An analysis of the daily mortality rate for the City of New Haven with respect to various meteorological conditions was made but again without significant results except for the broad seasonal effects. Taking the year as a whole, there were correlations of +0.27 with total positive ions, —0.25 with temperature and +0.17 with temperature change but these figures merely mean that the death-rate is higher in winter than in summer.

It is, of course, quite possible that with a larger body of data and for a longer period of time some day-by-day relationships might be apparent, but with the material at our disposal the results were entirely negligible.

Summary of Conclusions

1. The number of total ions in the air of a city street in New Haven is higher than in the adjacent countryside. It rises in winter and falls in summer. It rises with southerly (sea) winds. Total ion counts are highest in cool, moist and relatively windless days.

2. Judgment with regard to the pleasantness of outdoor atmospheric conditions, as evidenced by the vote of a group of some 80 persons taken each morning for a year, is chiefly influenced by sunshine (correlation +0.8), next by relative humidity (correlation —0.6), and total ions (correlation —0.3). When other relevant factors are held constant, these correlations remain significant and they are apparently independent factors directly affecting the sensation of pleasureableness connected with outdoor atmospheric conditions. The influence of ions is not related to electrical charge, since positive and negative ions show identical results. The authors are inclined to explain it as the result of the effect of those gaseous products of combustion (in houses and automobiles) which make up a large proportion of the ions in a city environment.

DISCUSSION

Mr. Haas: Were these data taken at all seasons of the year?

L. P. Herrington: Yes, the data covered a 15-month period which, of course, included the space of one seasonal year.

John Howatt: Dr. Hill, have you studied this paper enough to give us the benefit of your opinion?

Dr. E. V. Hill: No, I have not. There is one thought that occurred to me as I listened to the reading of the paper this morning. It appears to me that the investigators started out with the assumption that there is something wholly desirable and pleasant in the outdoor air, then attempted by this survey to find out what this desirable quality might be. If I am correct in that assumption it seems to me that the assumption is faulty, because year by year we live more and more indoors, and year by year our span of life increases, and our general condition improves. Hence it would seem to me that the best air conditions are indoors, and we should find out why they are not outdoors. The indoors, which we are coming more and more to use, and becoming more healthy as we do so, is really what we should study.

J. N. Hadjisky: The author indicated that there was an increase in comfort with a rise of barometric pressure.

Will the author make a distinction as to the amount of the rise of barometric pressure that is contributed by the rise of vapor pressure due to higher dew point and, how will he explain the fact that if one climbs up on the mountains where the barometric pressure is lower, one begins to feel better, at least I do.

W. A. Danielson: I wonder if the fact that we take more exercise outdoors is

really not the main difference between indoor comfort and outdoor comfort conditions. This might explain what the last speaker said. The higher he gets the more he has climbed and the more exercise he has obtained.

MR. HADJISKY: The case which I referred to was accomplished by a rise in elevation due to a train ride.

B. R. ROGERS[6]: I have been interested in the study of the physical force, "cold," as it affects the surface of the body, since 1907, and I believe there lies within your hands, you as physicists rather than the medical bacteriologists, the prevention of colds, influenza and pneumonia, by understanding the effect of cold on the surface of the body, driving the blood into the internal organs and producing inflammation.

MR. HERRINGTON: With reference to the first comment dealing with the nature of the assumptions which might have underlain this particular piece of work, we, of course, are not committed to the opinion that the outdoors is to be preferred to the manufactured condition under all circumstances, as might be well illustrated by our last week here in Chicago. We aimed simply to determine from actual judgments over a long period of time, what the relation is between the degree of pleasantness experienced outdoors and the physical weather factors which can be measured. It might be that the type of stimulation which is available in the outdoors produces certain desirable reactions, and that that situation cannot be reproduced indoors.

However, I see no particular reason why a collateral research of this general type, which admittedly is seeking suggestions, should not operate on the tentative assumption that factors determining pleasantness out-of-doors might be profitably considered in air conditioning. I think there have been assumptions in research work which were probably less well grounded than the assumption that the outdoors has something to it that is desirable from the weather standpoint.

With reference to the second question as to the relation between barometric change and temperature factors, or the other weather factors, these changes have been partialed out by a statistical method which uses the inter-correlations in the reported table which gives the interrelations of about 10 weather factors.

With reference to the matter of exercise, I am certain that any very definite conclusions as to the value of the outdoors as compared with the indoors, must be tremendously influenced by our habits of work and recreation. The outdoors obviously is the place where a great many people enjoy themselves, and the indoors is a place where a great many people do work which is, perhaps, of a less attractive nature, and all of us must react to these factors which are in part psychological rather than physical.

F. W. LEGLER: Most people assume that the air in the country is better than the air in the city, and I assume that these tests were all made near a large city. I make that assumption even though I heard Dr. Hill say at one time the air is better in the country because the farmers sleep with their windows closed. Why weren't the tests made in the country instead of the city?

MR. HERRINGTON: One of the chief reasons why they were made in the city rather than in the country was the fact that it would have been difficult to find a Security Insurance Company of New Haven, or a Bell Telephone Company, in the great outdoors. The collection of information of this general type is obviously simple in a mechanical way, but from a psychological standpoint, hard to get, for it demands the cooperation of 100 people for well over a year. It is practically impossible to get it except through stable groups of business people who are to be found day after day in the same location.

I should be very much interested to see what the reactions of 100 middle-western farmers might be to this particular questionnaire.

[6] Veterinarian, Chicago, Ill.

No. 1031

VENTILATION REQUIREMENTS

By C. P. Yaglou,* E. C. Riley ** (*MEMBERS*), and D. I. Coggins ** (*NON-MEMBER*), Boston, Mass.

This paper is the result of research sponsored by the American Society of Heating and Ventilating Engineers in cooperation with the School of Public Health, Harvard University

IN a previous paper [1] Lehmberg and his co-workers outlined a technic for studying ventilation requirements from the standpoint of body odors. Their work was purely fundamental, dealing largely with the development and evaluation of a suitable scale for judging intensity of body odors, and application of this scale to a laboratory experiment for studying the factors affecting odor intensity in a confined space.

The work to be described here is an elaboration of Lehmberg's preliminary experiments. The object was to study the general problem of ventilation odors under normal conditions, comparable to those in schoolrooms, offices, homes and the like with the possibility of establishing ventilation requirements for various groups of individuals, including grade school children and adults, under representative winter and summer conditions. Three methods of odor control were studied dealing with personal sanitation, ventilation, and air washing.

Principles of Olfaction

It is often stated in the literature, and rightly so, that the human nose when properly utilized affords a better criterion of the quality of air in occupied rooms than any of the known physical or chemical tests. Intelligent use of the sense of smell in ventilation requires a knowledge of its functions and limitations, a brief outline of which is given here.

The organs of smell are situated at the upper part of the nasal cavities. one in each cavity. The essential part of the organ is a delicate mucous membrane, the so-called olfactory epithelium, which covers the upper turbinal bone and

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[1] A Laboratory Study of Minimum Ventilation Requirements: Ventilation Box Experiments, W. H. Lehmberg, A. D. Brandt and Kenneth Morse, A. S. H. V. E. Transactions, Vol. 41, 1935.

Presented at the 42nd Annual Meeting of the American Society of Heating and Ventilating Engineers, Chicago, Ill., January, 1936, by C. P. Yaglou.

adjacent portion of the septum (see Fig. 1), an area only about 250 sq mm.[2] The membrane is yellowish in color and is innervated by filaments from the olfactory nerve. These filaments, or olfactory hairs, are believed to be lipoid in character and they constitute the true receptive elements of the organs of smell. They are bathed in a mucous secretion from numerous olfactory glands in the epithelium. In addition to olfactory cells, there are free nerve endings in the olfactory epithelium sensitive to irritants such as ammonia or sulfuric acid, but these are believed to have little or nothing to do with true odors.

It is now generally recognized that the olfactory organs are normally stimulated by molecules of odorous substances in the inspired air diffusing to the olfactory membrane where they must enter into solution with the mucous coating of the membrane before arousing the sense of smell by chemical action on the olfactory hairs.

In ordinary respiration, the course of air passing through the nasal cavity does not extend up high enough to strike the olfactory membrane, but the odoriferous particles reach the membrane by diffusion which gradually changes the air in the olfactory cleft. Sniffing appears to be the most effective and quickest way for the full perception of odors, as it directs the air stream upwards toward the olfactory areas.

Man's sense of smell is normally aroused by inconceivably small concentrations of odoriferous substances. The sensitivity varies in different individuals. In a very recent review on odors and odor control, Pierce[3] reports that $\frac{1}{2,000,000}$ of a milligram of oil of rose per cubic centimeter of air can be distinctly smelled. Mercaptan, a liquid with a penetrating garlic odor, can be smelled in much smaller concentrations. The odor of butyric acid, a constituent of the body odor complex, can be detected in concentrations of 0.000009 miligrams per cubic centimeter of air.

The olfactory organs are quickly and easily fatigued if the exciting stimulus continues, although they can perceive the sudden appearance of new odors. The occupants of a crowded and poorly ventilated room are not capable of recognizing body odors which are very apparent or even intolerable to a newcomer. Breathing fresh air restores the sensitivity. Therefore, in order to get a good impression of ventilation conditions in a room, one should pass quickly from clean outdoor air to the room to be tested. One or two sniffs produce the strongest sensations of body odor, after which the sensation diminishes and soon ceases altogether.

Recovery of excitability is apparently likewise rapid, although it varies to some extent with the concentration and length of exposure to odorous air. In our experience with body odors, the breathing of fresh air from 5 to 10 min restores full excitability after exposure of 3 to 4 hours in crowded and poorly ventilated rooms.

For best results the nose must be neither too dry nor too moist. Persons with colds are unable to smell odors. It is possible by mixing odoriferous sub-

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[2] Starling's Principles of Human Physiology, Lea and Febiger, Philadelphia, 1930.

[3] Odors and Odor Control, W. MacL. Pierce, A thesis submitted to the Department of Industrial Hygiene, Harvard School of Public Health for the Degree of Master of Science, January, 1935.

stances in certain proportions to annul or mask their effect on the olfactory organs. Likewise, in the presence of a strong odor, one cannot, as a rule, detect a weak odor simultaneously present in the air.

The unit of odor intensity is the olfactory threshold which itself is the smallest amount of an odorous substance required to stimulate the olfactory nerves. Different odors have different threshold values, and the threshold value of an odor expressed in grams per cubic centimeter is called an olfacty.

The intensity of odor perceived by the sense of smell does not vary in proportion to the concentration but approximately in proportion to the logarithm of the concentration. The variation is according to the law of physiological stimuli in general as expressed by the Weber-Fechner law,[4] namely, Sensation = K log. of stimulus-intensity.

SOURCES OF ODORS IN OCCUPIED ROOMS

Odors in living rooms come mostly from the occupants themselves. Odors from furniture, etc., are not, as a rule, conspicuous in the presence of body odors, although they may be accentuated in a warm or moist atmosphere. The

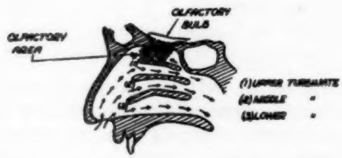

FIG. 1. SECTION THROUGH RIGHT NASAL CAVITY SHOWING OLFACTORY REGION AND DIRECTION OF AIR CURRENTS DURING INSPIRATION.

sources of body odors are numerous; foul breath, decaying teeth, sweat and sebacious secretions, especially when personal hygiene is deficient, gases from the digestive tract, and decomposition of matter from the skin and clothes are all contributory more or less, depending upon the socio-economic status of the occupants. Various gases, CH_4, CO_2, H, N, and H_2S have been detected[5] in intestinal gases eliminated from the rectum, and they may be tainted with the disagreeable odor of skatol and indol, which are normally present in the lower bowel.

Even healthy and clean persons freshly after a bath gave off in our experiments an appreciable amount of odor, which is apparently a normal waste product arising from metabolic processes and decomposition of matter in the skin and clothing.

Such odors are not, as a rule, known to be harmful, but they certainly induce a feeling of stuffiness and discomfort to anyone coming in from outside. The occupants themselves may not be conscious of the odor but they seem to be

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[4] On the determination of Odors and Tastes in Water, G. M. Fair, *Journal New Eng. Water Works Assn.*, Vol. 47 (Sept., 1933).

[5] The Health of the Industrial Worker, E. L. Collis and Major Greenwood, P. Blakiston's Son & Co., Philadelphia, 1921.

capable of detecting stuffiness or lack of freshness in the air. Sensitive persons are occasionally affected in a pathological way by sitting in such rooms. The untoward effects of body odors on appetite for food and performance of physical work were studied extensively by Winslow and Palmer in the laboratories of the New York State Commission on Ventilation.[6]

Despite the opposing schools of thought concerning the health aspects of body odors, the general agreement is that they should be controlled preferably by personal habits of cleanliness, or by ventilation and air conditioning, as it is difficult to control the source of odor itself. According to the latest views the air of occupied rooms should give a favorable impression on entering, taking into consideration such factors as odors, freshness, temperature, humidity, drafts and other factors affecting the senses. The present paper is largely concerned with such primary, as well as secondary sense impressions, which were taken as a criterion of ventilation requirements.

EQUIPMENT

The experiments were carried out in two adjoining rooms, on the northwest side of the building. The rooms were approximately identical, having a floor area of 155 sq ft, a ceiling height of 9 ft 2½ in., a window area of 20 sq ft and a net air space of 1410 cu ft approximately. The windows were weather-stripped and all cracks carefully sealed with adhesive tape.

One of the rooms, hereafter referred to as the experimental room, was occupied by the subjects who constituted the source of odor production. The other room served as a control room for the judges who estimated the odor intensity in the experimental room.

A small tight door was cut in the common partition to allow direct passage of the judges from the control room to the experimental room in order to judge the odor intensity and air quality. Separate air conditioning units, installed in the corridor, kept the two rooms at approximately the same temperature and humidity in any given test, so that the only variable factors in the two rooms were the outdoor air supply and the number and type of occupants.

One of the air conditioners was equipped with a centrifugal humidifier capable of fully saturating the air passing through it under the experimental conditions. The other one was of the conventional spray-dehumidifier type. The rated capacity of both was 1000 cfm, but the orifices employed for measuring airflow reduced it to about 500 cfm, which was ample for the purpose of the experiments. The air was introduced to the rooms near the ceiling through 14 in. round ducts running along the entire length of the rooms and fitted with splitters. The ducts were perforated over half of the periphery with a multitude of holes ½ in. in diameter and 2½ in. on centers, facing toward the ceiling. The recirculated air was withdrawn at floor level through a 10 in. round duct. The exhaust air was allowed to escape to the corridor through sensitive check louvres attached near the bottom of the doors.

Accurate measurements of the total air supplied to the rooms, the amount recirculated, and that taken from out of doors, were made by means of thin-

[6] Ventilation. Report N. Y. State Commission on Ventilation. E. P. Dutton & Co., New York, 1923.

plate orifices designed in accordance with the A. S. M. E. standards[7] and checked against a calibrated venturi meter. Control of the air flow was by means of variable speed motors and different size orifices.

Dry- and wet-bulb temperatures were measured by means of aspirating psychrometers, and the air movement by means of kata-thermometers, or globe anemometers.[8] Measurements of carbon dioxide were made by means of a 20 cc modified Haldane gas analysis apparatus[9] for CO_2 only.

EXPERIMENTAL PROCEDURE

Starting with a more or less definite plan, the procedure had to be modified during the first few experiments in order to allow for unforeseen conditions. During the preliminary part of the study, the outdoor air supply to the control room was fixed at the usual standard of 30 cfm per person with no recirculation. It was soon found out, however, when we began to study the influence of air space, that the odor strength in the control room was often higher than that in the experiment room. This upset the comparisons as it is physiologically impossible to judge odor strength by comparing an odor of standard intensity with one of lower intensity. The difficulty was overcome by increasing the outdoor air supply to 50 cfm per person and limiting the number of judges in the control room at any one time to 3, usually 2. This control standard was based on the minimum outdoor air supply required to keep the odor intensity in the control room approximately at the olfactory threshold, by comparison with clean outdoor air. The departure from the original plan is a happy one, as the olfactory threshold is a unit of odor intensity, and a much more rational base line than the arbitrary 30 cfm standard. The situation is analogous to our standards of sound or noise, in which the base line is likewise the threshold of hearing, that is a barely audible sound.

The experimental room was occupied by 3, 7, and 14 subjects in different series of experiments, so as to obtain 3 different floor areas per person, *i.e.*, 11, 22, and 52 sq ft and 3 different air spaces 100, 200 and 470 cu ft approximately. The air flow in the experimental room was varied from about 2 to 30 cfm per person in different experiments. In one series the total air supply remained constant at 30 cfm per person but the amount taken from out of doors was varied from 2 to 30 cfm. In another series only outdoor air was circulated through the experimental room. In a third series, the mixture of outdoor and recirculated air was washed, cooled, humidified or dehumidified in order to determine the effect of these processes on odor removal and on minimum ventilation requirements.

Keeping the air flow in the control room at 50 cfm per person in all experiments and the temperature and humidity approximately the same as that prevailing in the experimental room, the judges passed one at a time from the control to the experimental room, once every hour or so, recording the strength

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[7] Fluid meters and their application. Report of Comm. on Fluid Meters. Research Publication, *Am. Soc. Mech. Engrs.*, 3rd ed., 1931.
[8] A new instrument to be described elsewhere.
[9] Methods of Air Analysis, J. S. Haldane and J. I. Graham, Charles Griffin & Co., Ltd., London, 1935.

of body odor immediately after the change, according to the scale in Table 1. The agreement between judges was usually within ±½ point on the scale, as in Lehmberg's work, once they have become familiar with the scale.

Altogether 60 men and women with more or less normal sense of smell served as judges. They were drawn from employees of the school and graduate students. Two of the judges devoted their whole time to the tests. The others usually from 8 to 15 in each test were called in when needed, and after a short stay in the control room, they passed to the experimental room to smell the air, and were then released. All records were kept confidential by two men who ran the tests.

As a rule each test consumed a whole morning (9-12:30) or a whole afternoon (1:30-5:00). In a few instances the duration was shorter or longer.

TABLE 1. SENSORY INTENSITY SCALE OF BODY ODOR

Odor Intensity Index	Characteristic Term	Qualification
0	None	No perceptible odor.
½	Threshold	Very faint, barely detectable by trained judges; usually imperceptible to untrained persons.
1	Definite	Readily detectable by all normal persons but not objectionable.
2	Moderate	Neither pleasant nor disagreeable. Little or no objection. Allowable limit in rooms.
3	Strong	Objectionable. Air regarded with disfavor.
4	Very strong	Forcible, disagreeable.
5	Overpowering	Nauseating.

The equilibrium time with respect to odor intensity varied from one to three hours, inversely with the amount of outdoor air introduced.

Before and after each test the rooms were thoroughly ventilated in order to get rid of residual odors, and the air conditioning apparatus was kept clean at all times.

In order to secure uniformity in results and at the same time cover a fair cross-section of socio-economic status, the subjects were divided in 5 groups as follows:

(a) Sedentary men and women of average socio-economic status (between 16 and 60 years old), including high school, college and medical students, office workers, teachers, housewives, etc. (total 177).

(b) Grade school children between 7 and 14 years old of average or balanced socio-economic status (total 62).

(c) Laborers, such as janitors and street workers (total 8).

(d) School children of the poorest class (total 7).

(e) School children of the better class (total 28).

Most of the subjects belonged to groups *a* and *b*. They were apparently healthy and had normal personal habits. Variations from the normal were

studied in a limited number of subjects classified under groups *c, d,* and *e*. Some of the children in groups *b, d,* and *e* were selected by the principal and nurse of a nearby school and sent to the tests with their teacher.

All subjects were paid adequately to keep them interested in the work. No attempt was made to control personal habits of hygiene or nutrition. However, the use of cosmetics, including face powder, had to be forbidden owing to a masking effect upon odors. Individual variations did occur, even among persons of apparently the same socio-economic status and with the same number of baths, but these were smoothed off by the use of a large number of subjects, as would be the case in most public buildings.

During the experiments the subjects were seated comfortably in arm chairs, and they read, wrote or played cards. The clothing was of the customary indoor type according to the season.

The age, height, sex and occupation were recorded routinely in all tests. The date of last bath was also recorded more or less regularly, but this record was optional in the case of women. The weight was accurately determined before and after each test. At the beginning, near the middle and at the end of each test the adult subjects recorded their impressions of comfort, humidity, air quality and odor, according to a simple plan explained to them at the beginning of the test. The children answered questions pertaining to comfort only.

In this way information was obtained dealing with both primary and secondary impressions of air quality under various conditions of ventilation and with representative groups of persons. The primary impressions were those of the judges; the secondary, those of the subjects after becoming adapted to the conditions of the tests.

RESULTS

The data presented in this progress report deal with more or less comfortable conditions of temperature and humidity in the winter and summer. Observations with high and low humidities and with temperatures near the upper and lower boundaries of the comfort zones are yet to be completed. In addition many other questions are now being studied, bearing on the subject matter and data under consideration, for presentation in a final paper.

Throughout this work during the past three years, advantage has been taken of the splendid opportunities offered for parallel *sideline* studies of problems dealing with seasonal variation of comfortable air conditions in large groups of children and adults, sudden temperature contrasts, insensible perspiration, carbon dioxide output, humidity and air freshness, etc., which will be published in the future as time permits.

The mass of data in the present paper is condensed to essential facts in Table 7, and the reader who is interested in facts alone may save much time by referring to this tabulation.

Intensity of Body Odor in Relation to Outdoor Air Supply

(a) Observations with sedentary subjects.

In Fig. 2 are shown the results of experiments with simple ventilation using healthy sedentary subjects over 16 years of age. The data are representative of winter conditions when the air is simply tempered and circulated through the occupied space with or without recirculation. Ordinates in Fig. 2 give the

average odor intensity recorded by the judges in each test under equilibrium conditions. Abscissae give the corresponding quantity of outdoor air supplied per person per minute. In most instances the total air supply was 30 cfm per person (black circles) part of which (X) was taken from outdoors and the remaining part ($30-X$) was recirculated. The white circles represent tests in which there was no recirculation.

The first thing to notice in Fig. 2 is that when the odor intensity as perceived by the sense of smell is plotted against the logarithm of the outdoor air supply

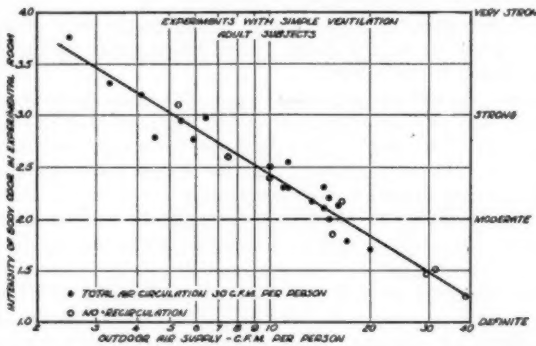

FIG. 2. OUTDOOR AIR SUPPLY IN RELATION TO ODOR INTENSITY. NET AIR SPACE PER PERSON, 200 CU FT

the relationship is linear. This is in accord with the Weber-Fechner law of physiological reactions in general, namely,

$$\text{Sensation} = K \text{ log. of stimulus}$$

and by inference

$$\text{Odor intensity index} = K \text{ log. of concentration of odor, or}$$

$$\text{“ \quad “ \quad “} = K \text{ log. } \frac{1}{\text{outdoor air supply}} \quad \text{and}$$

$$\frac{(\text{O.I.})_1}{(\text{O.I.})_2} = \frac{\text{log. cfm}_2}{\text{log. cfm}_1}$$

Translated into words, the strength of body odor perceived by the sense of smell on entering an occupied room from relatively clean air varies inversely as the log. of the outdoor air supply. The same fundamental law applies to the sense of hearing, and it may be of interest to call attention to the fact that our standards for noise and sound have been determined by similar subjective tests using the normal human ear for criterion.

It is seen in Fig. 2 that the body odor was very strong and disagreeable when the outdoor air supply per person was under 3 cfm, the strength decreasing arithmetically as the air supply increased logarithmically. The minimum air supply required to dilute the odor to the allowable intensity of 2 under the

given conditions (200 cu ft air space per person, no air conditioning) is about 16 cfm per person. With 30 cfm per person, the odor is still readily detectable but not objectionable.

Recirculation does not seem to affect the odor strength appreciably, as shown by the black and white circles in Fig. 2. In other words, from the standpoint of body odor, a room can be ventilated just as well with an outdoor air supply of 16 cfm per person as with a total supply of 30 cfm, about ½ of which is recirculated. Recirculation is often desirable for adequate distribution and temperature control, but one of the disadvantages is that it smells up the ducts, fans, etc., and unless the system is flushed frequently with clean air, higher air quantities will be needed to obtain satisfactory results.

Sex was not a factor in odor intensity and ventilation requirements in all instances in which female subjects used no perfumery of any sort, including face powder, before coming to the tests. Results of two series of experiments

TABLE 2. INTENSITY OF BODY ODOR WITH MEN AND WOMEN SUBJECTS UNDER COMPARABLE CONDITIONS

Subjects	Number of Tests	Total Number of Subjects	Air Space per Person Cu Ft	Average Outdoor Air Supply per Person Cfm	Average Odor Intensity
Women.....	4	23	200	15.8	2.05
Men........	5	35	200	15.9	2.00

with men and women subjects, under comparable conditions are shown in Table 2. Similar tests with grade school boys and girls showed no appreciable difference in odor intensity that could be attributed to sex.

(b) Observations with children of average class.

Grade school children between 7 and 14 years old have, apparently, an equation of their own. In spite of smaller body surface and lower total metabolism, they give off more odor than the adults and the air requirement is therefore considerably greater. The upper curve in Fig. 3 shows the results on a group of children of average or balanced socio-economic status. The conditions of the experiments were more or less comparable to those in schoolrooms with respect to air space, method of ventilation, activity, etc. The lower curve is reproduced from Fig. 2 for comparison with the results of adults under similar conditions. Whereas in the case of adults an outdoor air supply of 16 cfm per person was sufficient to take care of objectionable body odors, in children the air supply had to be increased to 21 cfm per child.

(c) Observations with adolescent children.

Adolescent boys and girls between the ages of 16 and 20 years yielded results closer to those of adults than those of grade school children. In 4 experiments, with a total of 28 adolescent boys and girls, using an air flow of about 10.5 cfm per person, the odor intensity averaged 2.5 ± 0.13, as compared with 2.4 for the adult subjects (See Fig. 2) and 2.7 for the grade school children (See

Fig. 3) under identical conditions. It may therefore be assumed that the requirements of adolescent children and adults are practically the same.

Air Space, Odor Intensity and Ventilation Requirements

(a) Observations with sedentary adult subjects.

The number of persons occupying a room appears to be a very important factor affecting odor intensity and air requirement as shown in Fig. 4. The 3 curves represent results with 3, 7, and 14 subjects in the room, corresponding to floor areas of 11, 22, and 52 sq ft per person and air spaces of 100, 200, and 470 cu ft respectively. The middle curve is reproduced from Fig. 2 without the points.

With 470 cu ft of air space per person, which is more or less representative of conditions in homes, uncrowded offices, and the like, the air requirement

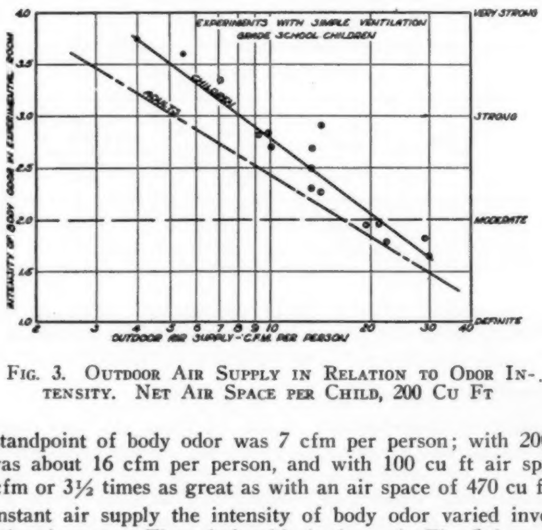

FIG. 3. OUTDOOR AIR SUPPLY IN RELATION TO ODOR INTENSITY. NET AIR SPACE PER CHILD, 200 CU FT

from the standpoint of body odor was 7 cfm per person; with 200 cu ft air space, it was about 16 cfm per person, and with 100 cu ft air space, it was almost 25 cfm or 3½ times as great as with an air space of 470 cu ft.

With constant air supply the intensity of body odor varied inversely with the log. of the air space. The relationship is shown in Fig. 5 from which it is possible to compute the ventilation requirements for intermediate densities of occupancy. There is no way to tell, except by actual tests, whether the values would hold with ceiling heights much greater than 10 ft, as for instance in theaters. The diffusion characteristic may be different there.

(b) Observations with children of average class.

Fig. 6 shows a similar picture with children as subjects. Comparison with Fig. 5 will reveal again higher odor intensities and air requirements in children than in adults in all 3 series of experiments. The ventilation requirements for children corresponding to the allowable odor intensity of 2 are

12 cfm per child with 470 cu ft air space
21 " " " " 200 " " " " and
29 " " " " 100 " " " "

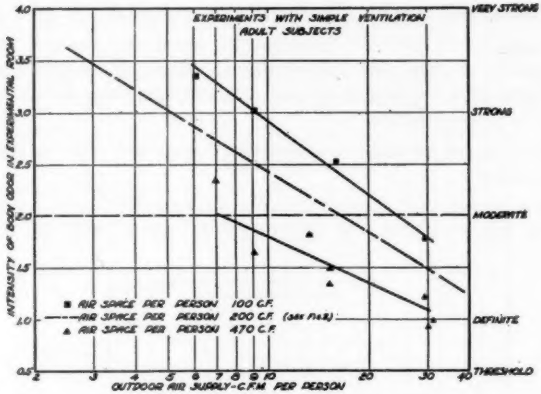

FIG. 4. INFLUENCE OF AIR SPACE ON ODOR INTENSITY. ADULT SUBJECTS

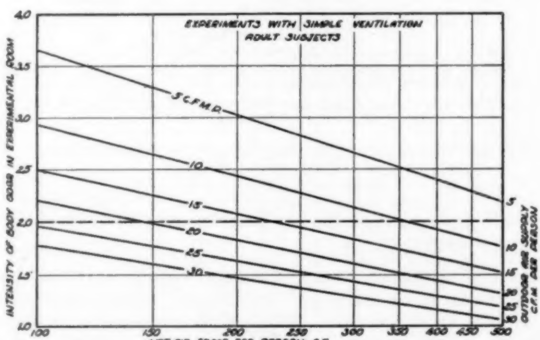

FIG. 5. CHART FOR COMPUTING VENTILATION REQUIREMENTS OF SEDENTARY ADULTS FROM THE STANDPOINT OF BODY ODOR

Example: Given an air space of 300 cu ft per person, follow vertical line through given air space until it meets the broken horizontal line passing through the allowable odor intensity of 2. The required outdoor air supply per person is 12 cfm obtained by interpolation.

as compared with 7, 16, and 25 cfm respectively for adults. Values for intermediate air spaces may be computed from Fig. 7.

The Influence of Air Conditioning Processes on Odor Intensity and Ventilation Requirements

The usual methods of washing, humidifying or cooling recirculated air were found to remove a considerable amount of odor, thus making it possible to

reduce the outdoor air supply. Three different arrangements were studied, as follows:

(a) mixture of outdoor and recirculated air passed through a conventional spray-type dehumidifier for cooling and dehumidifying the air of the experimental room in warm weather.

(b) mixture passed through a centrifugal humidifier, for humidifying the air in cold weather.

(c) mixture passed over a surface cooler through which cold water between 35 and 50 deg was circulated. The cooler used in these experiments was capable

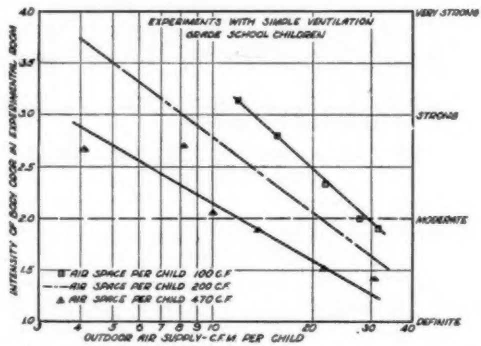

FIG. 6. INFLUENCE OF AIR SPACE ON ODOR INTENSITY. CHILDREN SUBJECTS

of lowering the temperature of the air passing over it (210 cfm) through a maximum of about 10 deg.

All 3 series of observations were carried out with 7 subjects in the experimental room (200 cu ft air space per person) and with a total air circulation of 210 cfm, or 30 cfm per person. The temperature range of the cold spray water and of the water circulated through the cooler was between 35 and 50 deg in different experiments. In the case of the centrifugal humidifier the range was 40-55 deg. In order to make the conditions uniform the spray water was changed before each test and the water tanks of the apparatus cleaned out periodically.

Fig. 8 shows the results. The top curve is again reproduced from Fig. 2 for comparison with the results of simple ventilation, that is with sprays and surface cooler off. The surface cooler absorbed the least amount of odor and the dehumidifier the most. The absorption by the centrifugal humidifier was but only slightly greater than that of the surface cooler. In all instances the surface of the cooler was wet with condensation which dripped to the humidifier tank below and overflowed to the sewer.

The outdoor air requirement was correspondingly reduced from 16 cfm per person with simple ventilation to about 13 cfm per person when the mixture of outdoor and recirculated air was passed through the centrifugal humidifier or

over the surface cooler, and to less than 4 cfm per person when the mixture was passed through the spray dehumidifier. In the last instance the odor intensity appears to be almost independent of the outdoor air supply. The performance of the dehumidifier may be more appreciated by calling attention

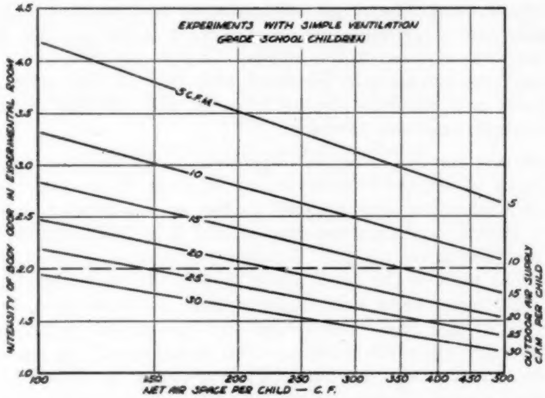

FIG. 7. CHART FOR COMPUTING VENTILATION REQUIREMENTS OF GRADE SCHOOL CHILDREN FROM THE STANDPOINT OF BODY ODOR. EXAMPLE IN THE USE OF CHART SHOWN IN FIG. 5

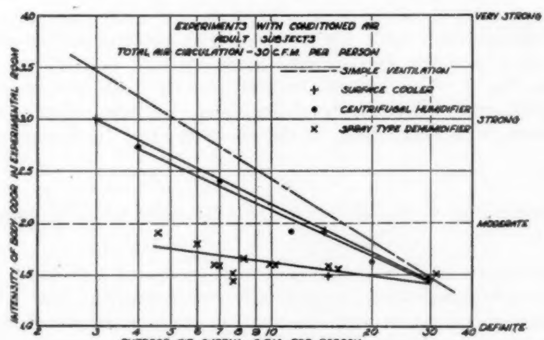

FIG. 8. ODOR REMOVING CAPACITY OF VARIOUS AIR CONDITIONING PROCESSES. NET AIR SPACE PER OCCUPANT, 200 CU FT

to the fact that the odor intensity cannot be lowered much below 1.5 when the total air circulation is limited to 30 cfm per person, even with a cleaner efficiency of 100 per cent (see Fig. 8). A possibility of olfactory delusion, rather than true odor absorption, cannot be entirely discounted.

While the results obtained with the surface cooler and centrifugal humidifier

will probably be realized in actual practice under similar operating conditions, the striking performance of the dehumidifier may be more or less limited by the following factors. The rated capacity of the dehumidifier spray system was about 15 gal per minute per 1000 cu ft of air circulated, but under the conditions of the experiments (210 cfm) the actual pumping rate corresponded in most instances to about 70 gpm per 1000 cu ft of air circulated. The unusually large pumping rate is probably not an important factor, as in a few tests in which the rate was reduced to the nominal 15 gpm per 1000 cu ft of air, the odor intensity was substantially identical with that of the higher rate. Undoubtedly there is a critical pumping rate, but the data are not sufficient at present to warrant a definite statement.

More important appears to be the cleanliness of the spray water, and water-storage capacity of the dehumidifier tank. In most of the experiments under consideration cold water was pumped to the sprays from a separate brine water-cooler having a storage capacity of 310 gal. The capacity of the dehumidifier tank itself was 10 gal. Another pump returned warm water from the dehumidifier tank to the brine water cooler. It is evident that the odor absorbed by the spray water was greatly diluted, owing to the unusually large water capacity of the brine cooler, and this presumably increased the odor removing capacity of the dehumidifier. The situation may be analogous in the case of large installations in which the spray water is cooled in a separate Baudelot cooler having a large storage capacity. Other factors at work appear to be the degree of flushing of the eliminator plates and the temperature of the spray water itself.

The importance of clean fresh water may be appreciated by the results of an experiment using old water from a previous test, after standing in the brine cooler and dehumidifier tank for 8 days. The outdoor air supply was 12.7 cfm per person, and the average odor intensity rose to 2.10, instead of 1.55 according to Fig. 8. This was the highest odor intensity ever recorded in any of our experiments with the spray dehumidifier. A few subjects in the room were conscious of a *musty* odor in the air even after remaining in the room 3½ hours.

Socio-Economic Factors in Relation to Odor Intensity, and Ventilation Requirements

The observations in Figs. 2 to 8 deal with groups of individuals of average or balanced socio-economic status and habits of personal hygiene. In a special series of experiments an attempt was made to study maximum variations from the average by using a limited number of subjects of the poorest and best class. With adult subjects the two extremes were represented by laborers (janitors and street workers), and medical students. In the case of school children the principal and nurse of a nearby school selected two groups, and the authors selected a third one from a different district, a group of children given daily baths and the best of care.

The results of these experiments are presented in Table 3. It will be noted that with equal ventilation rates, the laborers gave off considerably more odor than the medical students, and the ventilation requirements should therefore be greater. The computed air flow necessary to reduce the odor intensity to the

allowable limit of 2 is 23 cfm per laborer and 15.5 cfm per medical student, an excess of about 50 per cent. The air requirement of the medical students appears to be but slightly less than that of the whole group of sedentary adult subjects (See Fig. 2) probably because the majority of the subjects were medical students.

The probable ventilation requirements were computed in the following manner: If the three lines in Figs. 4 and 6 are extended to zero odor intensity,

TABLE 3. SOCIO-ECONOMIC STATUS, BATHS, ODOR INTENSITY, AND VENTILATION REQUIREMENTS

Experiments with Simple Ventilation

TYPE OF SUBJECTS	TOTAL NUMBER OF SUBJECTS	AIR SPACE PER PERSON CU FT	OUTDOOR AIR SUPPLY PER PERSON CFM	AVERAGE ODOR INTENSITY	COMPUTED VENTILATION REQUIREMENTS CFM PER PERSON	LAST BATH AVERAGE NUMBER OF DAYS
Laborers..........	7	200	7.3	3.29	24	7.2
Laborers..........	7	200	14.4	2.40	22	7.4
Average for Laborers	...	...	...	...	23	7.3
Medical Students..	7	200	7.5	2.60	15	1.3
Medical Students..	7	200	14.5	2.11	16	1.2
Average for Medical Students........	...	...	...	...	15.5	1.25
Average Sedentary Adult Subject (See Fig. 2)..........	..	200	...	...	16	2.2
Grade School Children of Poorest Class...........	7	200	22.0	2.83	38	8.0
Grade School Children of "Better Class"..........	14	200	20.9	1.87	18	3.0
Average Grade School Child (See Fig. 3)...........	..	200	...	...	21	4.2
Grade School Children of Best Class	14	100	16.4	2.34	22	0.8
Average Grade School Child (See Fig. 5)...........	..	100	...	...	29	4.9

they would all meet approximately on the horizontal scale at about 150 cfm. If now the experimental data of Table 3 are plotted on a chart similar to Figs. 4 or 6, a line joining each of the given points with the common point at 150 cfm would represent the probable relationship under the given conditions. The intersections of these lines with a horizontal line passing through the allowable odor intensity of 2 would then give the probable ventilation requirements under the various conditions.

In a similar manner the requirements of school children were found to vary from 18 to 38 cfm per child, according to socio-economic status, and this reflected objectively upon the bathing habits of individuals as can be seen in the last column of Table 3. To make sure about this point a group of children of average class and a group of medical students were tested separately, within about a day after a bath and complete change of underwear, and a week later, with no baths or change of underwear in between. The data are summarized in Table 4.

To begin with, the ventilation requirement of the children was about 10 per cent in excess of the medical students, but after about a week the difference

TABLE 4. BATHS, ODOR INTENSITY AND VENTILATION REQUIREMENTS

Experiments with Simple Ventilation

Type of Subjects	Air Space per Person 200 Cu Ft		
	Outdoor Air Supply Cfm per Person	Average Odor Intensity	Computed Ventilation Requirement Cfm per Person
Grade School Children			
0.5 Days after Bath and Complete Change of clothing	14.2	2.26	18
6.5 Days after Bath	14.3	2.90	29
Medical Students			
1.2 Days after Bath and Change of Underwear	14.5	2.11	16
7.0 Days after Bath	16.6	2.18	20

increased to 50 per cent approximately, owing presumably to the greater liability of children's clothing becoming soiled, and probably to other factors.

The significance of baths to ventilation requirements may be better appreciated by plotting the data of Tables 3 and 4 on a chart as in Fig. 9. The last two cases at the bottom of Table 3 have been omitted, as the air space differed from the others. The outstanding point in Fig. 9 is that, once the minimum ventilation requirements are fixed for any given conditions, the problem of body-odor control resolves itself to personal factors of hygiene and sanitation. Unfortunately the ventilating engineer has no control upon these factors; as a rule he has to accept the conditions as he finds them and design his system accordingly.

Another important point in Fig. 9 is that the proverbial weekly bath is not at all adequate from the ventilation standpoint, particularly in the case of schoolchildren. Two baths a week would help a great deal in solving the schoolroom odor problem. Unfortunately, there are homes of poor families with no bathing facilities at all. In a group of seven children of the poorest class, 3 frankly volunteered the information that they had to go to a club for their bath once every two weeks or so.

A worthwhile experiment from the standpoint of both economy and education would be to have grade schools in the poorer districts provided with baths and in this way treat the real cause with, perhaps, less expenditure of money, than would be the case with costly ventilation, which after all is temporarily corrective, not preventive. The mothers would then have more time to attend to the laundry of the children's clothing.

AIR QUALITY IN RELATION TO AIR SUPPLY AND ODOR INTENSITY

In discussing subjective impressions of air quality from the standpoint of air supply, two different viewpoints must be taken into consideration (*a*) that of the visitor upon entering a room from clean air and (*b*) that of the occupant

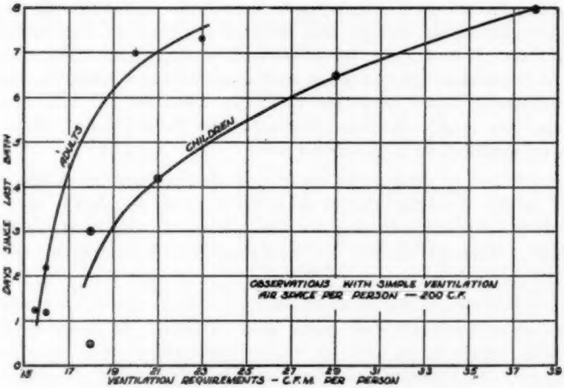

FIG. 9. EFFECT OF BATHS ON VENTILATION REQUIREMENTS. REQUIREMENTS BASED ON AN ALLOWABLE ODOR INTENSITY OF 2

after having become adapted to the conditions of the room. This suggests two ventilation standards both of which are valuable, the choice depending upon the nature of the ventilation problem.

In the present study the primary impressions of air quality were those of the judges; the secondary those of the subjects after exposures of 3½ hours to the air conditions under investigation. The primary or judges' impressions of air quality depended largely on odor intensity when both temperature and humidity conditions in the rooms were suitable.

The quality scale had five grades; *excellent, good, fair, poor,* and *bad,* and also a provision for a negative answer such as *cannot tell.* Air with an odor intensity of 1 was invariably qualified as good or good minus, odor intensity 2 corresponded to fair or fair to good, 2.5 was poor to fair, 3 bad, and 4 very bad. The correspondence was so close that it was found unnecessary to question the judges about air quality. Figs. 2 to 8 may therefore be utilized for illustrating approximately the relationship between primary impressions of air

quality and outdoor air supply, by substituting impressions of air quality for odor intensity on the vertical scale.

The subjects, on the other hand, could not, as a rule, detect the odor, except when there were 14 of them in the room. Even then however only about a third remarked about odors and the majority in this third were not at all certain whether it was body odor or odors of some other kind. The subjects' criterion of good air was a *freshness complex* perceived by the nose. The most popular reasons for poor air quality, under comfortable conditions of temperature and humidity, were variously described as *stuffiness, closeness, heaviness, lack of freshness,* etc.

Upon entering the room at the beginning of the tests the subjects generally agreed that the quality was good or excellent. The next record near the middle of the test showed a depreciation of quality, and the third record at the end of the test showed but minor changes from the second. Most of the observations on air quality came from college and medical students, an intelligent group of persons capable of analyzing their feelings and voting quite concordantly. Observations from other groups were on the whole less consistent, except under extreme conditions. They could tell definitely when the air was good or poor but as a rule they could not discriminate intermediate grades. Sometimes the answers were negative, such as *cannot tell.*

The quality of air in the control room was always good or excellent (excepting tests in which the temperature was too high or too low), and there was no depreciation of quality throughout the period of the tests. This is to be expected with a constant outdoor air supply of 50 cfm per person in all experiments.

In the classification of the data shown in Table 5, all observations in which the subjects were warm or cold have been excluded, in order to restrict the relationship, as much as possible, to factors pertaining to air quality and outdoor air supply. The results are shown graphically in Fig. 10 which was constructed by evaluating grades of air quality in terms of numbers.

It will be observed that with simple ventilation and with 200 cu ft air space per person, the air quality was poor to bad when the air supply was under 3 cfm per person, improving rapidly as the air supply increased to 15 cfm. Beyond this point further increases in the ventilation rate had little or no effect on air quality, from the standpoint of the occupants, and it will be recalled that from the standpoint of the judges the odor was not objectionable with airflows in excess of 16 cfm (Fig. 2). The general trend is similar, when the air space is reduced to 100 cu ft per person, but the air quality is inferior throughout, a situation which is consistent with the higher odor intensity reported by the judges (Fig. 4).

With an air space of 470 cu ft, on the other hand, air quality appears to be practically independent of air supply, in so far as the occupants are concerned, when the period of occupancy does not exceed 3½ hours. An outdoor air supply of 7 cfm per person was ample from the standpoint of both subjects (Fig. 10) and judges (Fig. 4). The same was more or less true when the air was cooled and dehumidified by passing through the dehumidifier as shown in Figs. 10 and 8.

Table 6 gives additional observations of air quality, those recorded by the

attendant or teacher in the children's experiments. These data are necessarily limited because each test yielded a single observation only, that of the attendant or teacher. In experiments with 3 children, observations of air quality had to be omitted as the children were unattended.

To check this apparent close correspondence between primary impressions of odor intensity and secondary impressions of air quality we plotted the values

TABLE 5. AIR QUALITY, OUTDOOR AIR SUPPLY, AND ODOR INTENSITY

Impressions of Subjects After Exposure of 3½ Hours to Comfortable Conditions of Temperature and Humidity

Outdoor Air Supply Cfm per Person	Mean Air Supply Cfm per Person	Mean Odor Intensity[a]	Total Number of Subjects	Air Quality Number of Subjects Recording					Mean Air Quality
				Excellent (5)	Good (4)	Fair (3)	Poor (2)	Bad (1)	
Simple Ventilation Tests. Air Space per Person 200 Cu Ft									
Under 3	2.5	3.75	7	..	..	1	3	3	1.7
3–5	3.7	3.21	25	..	1	18	5	1	2.8
5–7	5.7	3.08	33	..	3	22	8	..	2.9
7–10	8.7	2.91	14	..	5	8	1	..	3.3
10–15	13.9	2.20	42	1	29	12	..	..	3.7
15–17	15.8	2.02	27	4	16	7	..	..	3.9
29–31	29.7	1.54	15	3	10	2	..	..	4.0
38.7	38.7	1.25	6	..	6	..	..	..	4.0
Simple Ventilation Tests. Air Space per Person 470 Cu Ft									
7–9	8.1	2.00	6	..	4	2	..	..	3.7
13–16	14.6	1.55	9	..	6	3	..	..	3.9
29–32	30.3	1.11	6	..	5	1	..	..	3.8
Simple Ventilation Tests. Air Space per Person 100 Cu Ft									
6.1	6.1	3.37	14	..	..	4	8	2	2.1
16.0	16.0	2.52	14	..	6	5	3	..	3.2
29.4	29.4	1.78	14	2	5	7	..	..	3.6
Experiments with Spray Dehumidifier. Air Space per Person 200 Cu Ft									
4–7	6.1	1.72	24	..	14	10	..	..	3.6
7–10	8.4	1.56	26	..	14	12	..	..	3.5
10–15	12.5	1.59	14	..	11	3	..	..	3.8
31.2	31.2	1.51	7	2	3	2	..	..	4.0

[a] Impressions of judges upon entering room from relatively clean air of threshold odor intensity.

of these two variables, appearing in Tables 5 and 6, against one another, as shown in Fig. 11. It can be seen that the factors of air supply, air space, as well as those pertaining to individual differences and to air conditioning processes, almost disappear from the picture, indicating that an underlying factor in air quality, when both temperature and humidity are controlled, is the odoriferous organic matter given off by the human body. This is in spite of the fact that the subjects themselves could not smell the odor. Air flow, air

space, air conditioning processes, and personal sanitation, are apparently secondary factors affecting the concentration of the odoriferous matter.

In the Study of the N. Y. State Commission on Ventilation, Winslow and Palmer [10] arrived at a somewhat analogous conclusion from an entirely different angle. Quoting from their original report,—"these experiments seem to warrant the conclusion that there are substances in the air of an unventilated occupied room (even when temperature and humidity are controlled) which in some way and without producing conscious discomfort or detectable physiological symptoms diminish the appetite for food. The observed beneficial effects of fresh air may to some extent be connected with this phenomenon."

TABLE 6. AIR QUALITY, OUTDOOR AIR SUPPLY, AND ODOR INTENSITY

Impressions of Attendant or Teacher in Children's Experiments after Exposure of 3½ Hours
(Observations with Simple Ventilation)

Outdoor Air Supply Cfm per Person	Mean Air Supply Cfm per Person	Mean Odor Intensity	Number of Observations	Mean Air Quality
	Air Space per Child 200 Cu Ft			
Under 6	4.8	3.67	2	2.5 (Poor to Fair)
10–15	13.0	2.48	5	3.4 (Fair to Good)
19–23	21.1	1.91	4	3.8 (Good)
30	30.0	1.65	2	4.0 (Good)
	Air Space per Child 100 Cu Ft			
15.7	15.7	2.80	1	3.0 (Fair)
21.8	21.8	2.34	1	3.5 (Fair to Good)
31.5	31.5	1.90	1	4.0 (Good)

CARBON DIOXIDE IN RELATION TO OUTDOOR AIR SUPPLY AND ODOR INTENSITY

Since the time of Max von Pettenkofer, about 75 years ago, the carbon dioxide content of air in occupied rooms has been widely used as a measure of fresh air supply and a convenient yardstick for the degree of air vitiation by products of organic decomposition from the human body. The value of this index in ventilation work has often been questioned by many workers, and recently by Houghten [11] who showed that changes in the moisture content of air in occupied rooms constitute a more reliable and convenient index of air supply than CO_2 itself. Houghten did not, however, measure the actual CO_2 content of air in his experiments but he computed it from approximate metabolic relationships.

In the present study the actual CO_2 in the air was determined in most experiments near the end of the tests, at different stations inside the experimental

[10] Effect Upon Appetite of the Chemical Constituents of the Air in Occupied Rooms, C.-E. A. Winslow and G. T. Palmer, *Proc. Soc. Biol. & Med.*, vol. 12, p. 141 (1914-15).
[11] Indices of Air Change and Air Distribution, F. C. Houghten and J. L. Blackshaw, A. S. H. V. E. TRANSACTIONS, Vol. 39, 1933.

room or in the exhaust leading to the corridor. The relationship between CO_2 and outdoor air supply is given in Fig. 12. The curve was drawn from the well known formula

$$\text{Cfm per person} = \frac{100}{CO_2 - 3.5}$$

where 3.5 is approximately the average value of CO_2 in parts per 10,000 as found in the air entering the apparatus before mixing with recirculated air.

Although the theoretical curve roughly averages the experimental points, great discrepancies occur in the steep portion of the curve which happens to

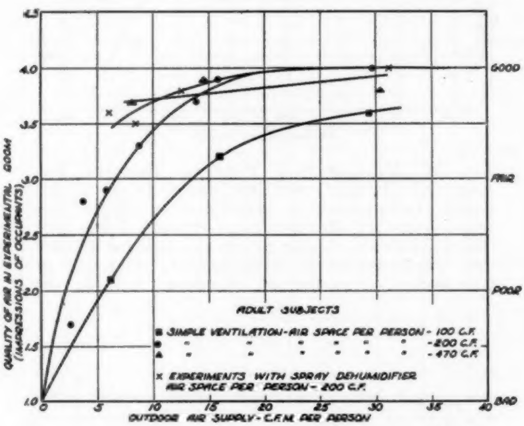

FIG. 10. AIR QUALITY IN RELATION TO OUTDOOR AIR SUPPLY UNDER VARIOUS CONDITIONS. IMPRESSIONS OF OCCUPANTS (SUBJECTS) AFTER EXPOSURES OF 3½ HOURS TO THE SPECIFIED CONDITIONS

be the practical range used in ventilation work. For example, a CO_2 of about 9 parts per 10,000 may mean an actual ventilation rate between 15 and 30 cfm per person, or a maximum error of 100 per cent, depending largely upon the amount of CO_2 given off by the particular group of occupants in the room. The curve is so steep for ventilation rates above 10 cfm, that even small individual variations in CO_2 output correspond to large variations in the air supply. The usual errors in determining CO_2 add to the discrepancies.

For airflows under 10 cfm per person, the discrepancy is not so great, but the computed air flow from the equation is almost always higher than the actual airflow, owing presumably to absorption of CO_2 by the walls, furniture, clothing, etc., when the concentration is high.

As an index of body odor CO_2 appears to be still worse, as can be gathered from the dispersion of the experimental points in Fig. 13. The reasons are quite obvious. Suppose, for instance, that a person takes no bath for a week or two, the odor intensity would be considerably increased (see Fig. 9) but

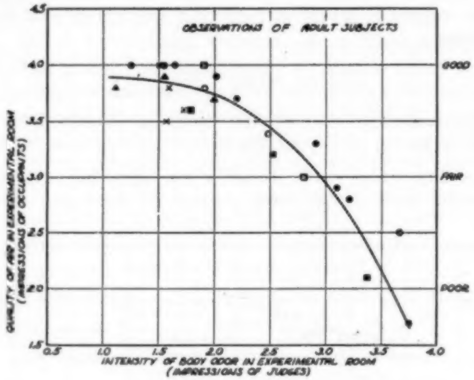

FIG. 11. RELATIONSHIP BETWEEN PRIMARY IMPRESSIONS (JUDGES') OF ODOR INTENSITY AND OCCUPANTS' IMPRESSIONS OF AIR QUALITY AFTER EXPOSURES OF 3½ HOURS. SOLID POINTS REPRESENT OBSERVATIONS OF ADULT SUBJECTS; HOLLOW POINTS ARE OBSERVATIONS OF ATTENDANT OR TEACHER IN CHILDREN'S EXPERIMENT. LEGEND SHOWN IN FIG. 12

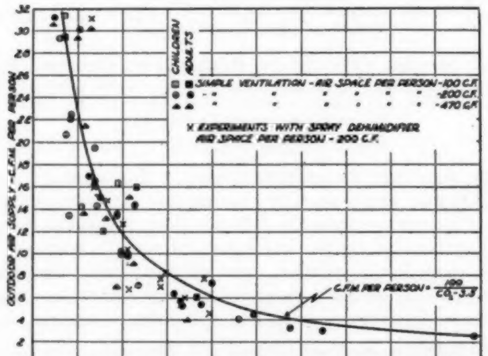

FIG. 12. CARBON DIOXIDE CONTENT OF AIR IN EXPERIMENTAL ROOM IN RELATION TO OUTDOOR AIR SUPPLY

the CO_2 output would barely be affected. Children, for example, in spite of their low CO_2 excretion give off more odor than the adults (Fig. 13) owing largely to differences in bathing habits and cleanliness of clothing, as has been related. On the other hand, when recirculated air is passed through an air-washer, the water removes a considerable portion of the odoriferous matter without affecting much the concentration of CO_2 (Fig. 13).

In the light of these and other similar data by the New York State Commission on Ventilation, and others, it is evident that a great deal of unjustified effort would be saved by discontinuing the usual measurements of CO_2 in ordinary ventilation work, except perhaps in instances in which the airflow is well

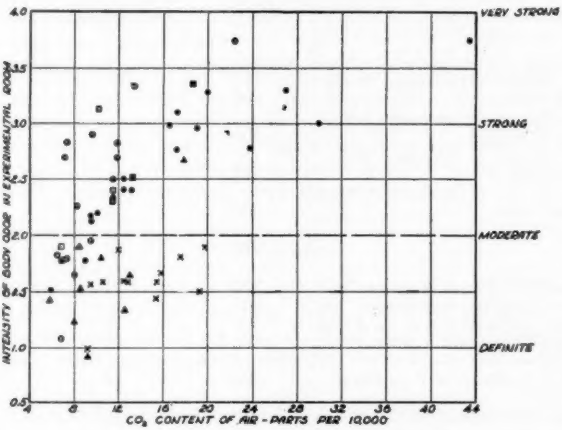

FIG. 13. CARBON DIOXIDE CONTENT OF AIR IN EXPERIMENTAL ROOM IN RELATION TO ODOR INTENSITY. FOR LEGEND SEE FIG. 12

under 10 cfm per person. Factors pertaining to air distribution can be studied much easier by variations of temperature and air movement from station to station than by variations of CO_2, as outlined in the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS GUIDE (1935, p. 51).

THE A. S. H. V. E. AND OTHER EARLIER STANDARDS

In Table 7 are summarized the ventilation requirements under various conditions as determined in the present study. The impossibility of fixing any single value that would apply under all conditions is evident. Each case would have to be considered on its own merits. The A. S. H. V. E. provisional standard of 10 cfm per person seems to be a more or less fair average value for adult persons where the air space per occupant is between 250 and 500 cu ft per occupant. It falls far too short in grade schools attended exclusively by children of poor districts, assuming that the results are representative, and some-

what too long when the air is cooled and dehumidified in a spray-type air conditioner.

Likewise the results of the present study are not necessarily contradictory to those of the old masters (De Chaumont, Parkes, Billings, and others). The

TABLE 7. SUMMARY OF MINIMUM OUTDOOR AIR REQUIREMENTS FOR VENTILATION UNDER VARIOUS CONDITIONS

(Provisional Values Subject to Revision upon Completion of Work)

TYPE OF OCCUPANTS	AIR SPACE PER PERSON CU FT	REQUIREMENTS BASED ON PRIMARY[a] IMPRESSIONS CFM PER PERSON	REQUIREMENTS BASED ON IMPRESSIONS OF OCCUPANTS[b] CFM PER PERSON
Heating Season with or without Recirculation. Air Not Conditioned.			
Sedentary Adults of Average Socio-Economic Status	100	25	23
Sedentary Adults of Average Socio-Economic Status	200	16	11
Sedentary Adults of Average Socio-Economic Status	300	12	..
Sedentary Adults of Average Socio-Economic Status	500	7	>5
Laborers	200	23	..
Grade School Children of Average Class	100	29	..
Grade School Children of Average Class	200	21	15
Grade School Children of Average Class	300	17	..
Grade School Children of Average Class	500	11	..
Grade School Children of Poor Class	200	38	..
Grade School Children of Better Class	200	18	..
Grade School Children of Best Class	100	22	..
Heating Season. Air Humidified by Means of Centrifugal Humidifier. Total Air Circulation 30 Cfm per Person.			
Sedentary Adults	200	12	..
Summer Season. Air Cooled and Dehumidified by Means of a Spray Dehumidifier. Total Air Circulation 30 Cfm per Person.			
Sedentary Adults	200	>4[c]	6[c]

[a] Impressions upon entering room from relatively clean air at threshold odor intensity. Allowable odor intensity = 2. (For scale, see Table 1).
[b] Corresponding to an air quality of fair to good.
[c] Values provisionally restricted to the conditions of the tests.

differences in the requirements between then (50 to 30 cfm per pupil) and now (Table 7) are accountable by differences in standards of personal sanitation, as Fig. 9 would seem to indicate.

In the light of the present study, two sets of requirements are suggested

as shown in Table 2 to serve two different purposes. In buildings with transient occupants, such as theaters, banks, restaurants, and the like, where the primary impression of the patrons upon entering is of considerable importance, the higher requirements would be better suited in spite of the higher cost. On the other hand, in homes, offices, schools, etc., the lower or secondary requirements may be more desirable from the economic standpoint. Lacking values in Table 7 will be completed soon and weak data will be strengthened and revised in accordance with additional work. The important omission of information on winter air conditioning will also be attended to.

It should be clearly understood that the requirements in Table 7 are average values determined by averaging impressions of a large group of persons. Individual variations will naturally occur; there will always be a few persons who would prefer somewhat higher standards as well as a few others who would be content with lower standards.

Given the specifications of minimum outdoor air requirements under various conditions, the problem would then resolve to recirculating a sufficient amount of air when needed in order to maintain proper temperature, humidity, and air movement. Temperature, in fact, is one of the most important factors in air quality and unless it is controlled the quality will suffer badly no matter what the outdoor air supply, particularly when the air is overheated.

SUMMARY

1. The outdoor air requirements for ventilation under comfortable condition of temperature and humidity have been determined from primary impressions of odor intensity upon entering occupied rooms and from impressions of the occupants themselves after exposures of 3½ hours to the conditions investigated.

2. Wide individual variation occurred in the amount of odor emitted by various groups of persons, according to socio-economic status, especially the bathing habits of individuals and cleanliness of clothing. Children, as a rule, gave off more odor than adults, and their bathing habits were deficient.

3. Even healthy clean persons freshly after a bath gave off an appreciable amount of odor which required from 15 to 18 cfm of outdoor air per person in order to dilute it to a concentration that was not objectionable to persons entering the room from relatively clean air. A week after a bath the ventilation requirement of children increased from 18 to 29 cfm, as compared with an increase of from 15 to 20 in the case of adults.

4. With a given group of occupants the intensity of body odor perceived upon entering a room from relatively clear air (air with threshold odor intensity) varied inversely with the logarithm of the quantity of outdoor air supplied and the logarithm of the air space allowed per person.

5. Untreated recirculated air in any amount had no effect on odor intensity or quality of air.

6. The usual processes of washing, humidifying, cooling, and dehumidifying recirculated air apparently removed a considerable amount of body odor and, under certain conditions, practically the maximum amount possible by the use of known processes.

7. Under comfortable conditions of temperature and humidity, both primary and secondary impressions of air quality were found to be related closely to the concentration of odor in the air, despite the fact that the occupants themselves could not smell the odor.

8. Based upon these impressions, two sets of ventilation requirements were derived for various groups of individuals under various conditions as summarized in Table 7. The impossibility of fixing any single standard that would apply under all conditions is clearly evident. Each case, it would seem, must be considered on its own merits.

9. The concentration of CO_2 in the air of occupied rooms proved to be an unreliable index of ventilation, from the standpoint of both outdoor air supply and odor intensity.

Acknowledgment

Most of the preliminary painstaking work was done by Messrs. J. W. Buford, Gayle Priester, and Joseph Gessner, to whom the authors are greatly indebted.

DISCUSSION

J. D. Cassell: This paper is of particular interest to me as it strengthens the conclusions of the Pennsylvania State Committee appointed to study and report on rules to govern air supply to public school classrooms. At the start of our work the State Committee adopted the report of the Ventilation Standards Committee of this Society, by providing 10 cu ft of outside air, and 20 cu ft of recirculated air per occupant of room. These proportions were found insufficient to eliminate offensive odors; so after further investigation the Committee finally agreed on 15 cu ft of outside air and 15 cu ft recirculated air per minute per occupant of room. This final report has not been made public, probably due to a change in state administration.

I wish to congratulate Professor Yaglou, not only on this most thorough and enlightening paper, but also in the masterly manner in which he presented it. I hope to hear comments from other school men present. By the way, I am no longer a school man; there comes a time in some men's life when they just can't take it, and I am one.

N. W. Downes: This paper brings to mind an experience we had in Kansas City some years ago. We built a 30-room elementary school in the north end of the city for serving children such as Professor Yaglou referred to as coming from the more unfortunate class. The building was equipped with a warm blast fan system, with air washer, automatic temperature and humidity control and so dampered that not only outside air in toto could be used but air could be recirculated on a basis of one-third, two-thirds or in toto, the air supplied remaining constant at 30 cfm per pupil. We soon found we were getting no place on a recirculation basis on account of odors which were at times nauseating. Even the use of outside air in toto failed to eliminate the objections.

The Principal on investigation soon found that a large number of these youngsters were sent to school by their parents literally sewed up for the winter. He immediately took it upon himself to change the situation by requiring each child to take a warm shower bath once a week at the school, although encountering vigorous opposition from some of the parents. The objectionable odors soon disappeared as did also the objections from the parents. Afterwards it was found possible to recirculate through the air washer as much as two-thirds of the total without objectionable odors.

G. P. Ellis: In Pittsburgh we continue to use 30 cfm per pupil. However, in schools now being built and in operation since September 1, we have equipment whereby we can recirculate 50 per cent, but the control of that equipment and the capacity of the equipment is such that we can adjust it up to 30 cfm whenever we feel like it, and feel it is necessary to do so.

J. N. Hadjisky: I do not represent Detroit officially, but I know the conditions there as they existed during the period 1920 to 1928. The practice during that time was to circulate 30 cfm per person. The percentage of recirculated air is dependent on outside temperature. A fair average for the cold winter period was 50 per cent. In some of the schools in the outlying subdivisions, outside of the city limits, it was a common practice, in very cold weather, to close the fresh air dampers altogether. Naturally the odors in the rooms were very noticeable.

I made a test once in a small school, with a ventilating system handling 30,000 cfm. If the fresh air dampers were closed for half hour, the odors become very noticeable, in spite of the continual washing of the 100 per cent recirculated air.

I should like to ask Professor Yaglou a question in connection with his tests when expanded surface coils were used. Some years ago when I was testing some air cooling units, I noticed that the units having direct expansion coils and adjusted to keep the coils just frost covered, the unit did not keep the air in the room as fresh as when the room was cooled with the unit using cold water as the cooling medium in the coils. In this last case the wetted surface of the coil seemed to have a better cleansing effect upon the recirculated air than the dry, frost covered coil, of the direct expansion type. The condensation which was collected, showed by its color that it had a great deal of dirt and color.

Mr. Shepherd: I would like to ask Professor Yaglou if the water used in the spray type dehumidifier was fresh or had been used long enough to become saturated relative to the odors.

S. R. Lewis: To say that 50 per cent or 20 per cent of the air delivered to a school room is outside air and that the balance is recirculated air means very little in practical service because ducts and walls are so porous and leaky that no exact or consistent measurement is possible. The recirculated part of the air may be badly contaminated or it may be uncontaminated even though it has traversed the room.

The distribution of the air within the room so that each occupant shall receive his share of the entering air has a very great deal to do with the problem. Suppose that actually 25 per cent outside air and 75 per cent recirculated air are introduced. If each person receives his share there may be no evidence of odor or of improper heat removal. However, there are the ever-present conditions of poor distribution, dead corners, down drafts from cool windows, and up drafts from warm surfaces, which create turmoil and which render the equal distribution of the air a complex matter.

There is in my experience neither odor trouble nor temperature control trouble attending partial recirculation but I doubt whether the exact amount of contact with contaminated air can be measured or expressed as a percentage unless the room and ducts are bottle-tight.

Member: The water used in the air washers of the Chicago Public School systems is supplied continuously fresh. A ¾-in. pipe runs into the tank while the air is being recirculated, and a certain level is maintained for overflow. A great number of our troubles originate with odors in connection with these systems.

J. J. Aeberly: It is not gracious at this time for a committee member to make a critical analysis of the report his chairman submits. For this reason I would like to make only a few friendly remarks.

This is a very significant paper. The air conditioning engineer has recognized

the importance of the need for control of odors in ventilating systems. I have had occasion to investigate a number of air conditioning jobs a long time after the engineer had completed his work. Without exception, the problem today in these systems is the elimination of odors.

I would like to say, Professor Yaglou, that it would be well to make some intensive study of the effect of air-washers, particularly those systems where dew point control is used and similar controls where the water is recirculated. It may be that under these conditions the effectiveness of air-washers will not be as great, although I think your findings are substantially correct for non-recirculated water. I think you will all agree with me that Professor Yaglou did a mighty fine piece of work.

W. H. DRISCOLL: I think this is one of the finest contributions to the literature of this Society we have had in many a day. Those of us who worked on the Ventilation Standards Committee, know how difficult was the problem of arriving at a decision as to what we should set up as the minimum outdoor air requirement.

I think that Committee worked for well on to two years, possibly more than two years, before it had its report in final shape for submission to this Society. I venture to say that nine-tenths of the time we spent in debate and discussion was spent in a consideration of the question of air volume, and outdoor air requirements.

When we finally decided that we would take 10 cfm per person as the minimum, it was a sheer compromise, merely an attempt to finish the work of the Committee and get the report before the Society. There was a difference of opinion as to whether the 30 cu ft that have been set up as a standard since time immemorial should be adopted, or whether no cubic feet, for which there was very aggressive support, not necessarily within the Committee but from outside of the Committee, on the theory that no scientific studies had ever been made to support the necessity for the introduction of any outdoor air as a ventilation requirement.

The work that Professor Yaglou has done has helped materially to bring out the necessity of some outdoor air. I am quite sure that it will galvanize into activity the Committee on Ventilation Standards which has been inactive for some time, primarily because we wanted the standards that we had set forth to circulate, to be tried out, to get the reactions that might come from them. I think that Professor Yaglou deserves our gratitude and I want at this moment to pay my personal tribute to him for what he has done, and the great work he has always done for this Society. I have every confidence in the findings and conclusions arrived at by him in the investigations he makes. I know something of his work. I keep in close contact with him, and I feel that he has given us something to work on that we have never had before in the history of ventilation.

MEMBER: I am particularly interested in the length of time these occupants were exposed. It would have a direct bearing on the results.

MR. HAAS: I was interested in the tremendous effect the per cubic foot of space of occupant had on the total air circulated. In totaling the values given in this paper it will be noted that six times as much air per hour is required, if it is circulated in a small space, as if one sixth of the amount of additional space in the volume of the room is provided. What difference does it make whether the air is in the room, or circulated?

E. V. FINERAN: It appears to me on the air washer chart that the curve is drawn with the wrong slope. This curve indicates that a bad odor condition would never be reached, yet if it is extrapolated on the other end, an infinite quantity of outside air is required to approach a basic level.

J. H. VAN ALSBURG: This is a fine paper because it presents basic information, which has a direct and practical application. I have just one suggestion, namely, I hope that the Committee and the investigations will continue to cover one more series of conditions, such as the requirements for bed rooms in hotels, in homes and in railroad sleeping cars.

Dr. E. V. Hill: I cannot let the opportunity go by to have my say on this subject. I want to agree with Mr. Aeberly about the importance of the subject of odors. It is a thing I seldom do is to agree with Mr. Aeberly, but I will in this case.

The paper is good; it is giving us information that we need, but it seems to me that we could get more fundamental information if we approached the problem in a little different way.

If I want to know how much air is required to remove the odors from a human being, I would test the amount of odors given off by a human being nude, without any clothes on. Then I would put the clothing in a cabinet and see how much air was required to remove odors from the clothes. In that way you would have some fundamental information.

I did not hear Professor Yaglou mention the character of the clothing worn by the different subjects, and it seems to me that this is a very important thing. I have expressed myself on this subject before, and some one always gets up and says, "Well, we don't have our children nude in school, or people nude in theaters. Why test nude subjects?" It is the same old subject. It is the same thing that applies in our work on the effect of temperature, or whatever the temperature is that produces comfort. We want to know fundamentally how the air affects the human body. We want to know in this particular case how much air is required to remove the odors given off by normal healthy clean human bodies.

Professor Yaglou indicated the fallacy in the presentation by showing the difference in the air requirement after the children were washed, so it seems to me that we should get away from this silly old idea we have got to experiment with normally clothed people. The doctor doesn't examine a patient with the clothing on. He would be laughed out of the profession. Why should we do physiological experimental work with the subjects clothed. Let us get down to a scientific basis, and do our experimental work in this where we are dealing with the physiological effects of air on nude subjects, and find out how clothing modifies the picture.

Mr. Driscoll: Somebody always gets up and disagrees with Dr. Hill so I don't want to disappoint him. I am concerned primarily with people who are clothed, and the circumstances under which we as engineers have to treat those people.

W. F. Christman: This discussion refers to the idea that we have a bare room, with no draperies, rugs, or things of that nature. What effect on the recommendation that Professor Yaglou makes has the question of draperies or rugs or any odor-giving device of that nature? All of the discussion up to this time seems to be concerned with the odor question. I would like to know if there is any effect on the bacteriological count with regard to the amount of outside air introduced.

Mr. Aeberly: Although this paper clearly indicates that the results obtained with air-washers are based on non-recirculation of the wash water, this fact should be emphasized because in practice similar results are ofttimes expected, but are not obtained when wash waters are recirculated.

Professor Yaglou: I agree with you, Mr. Aeberly; in our experiments the spray water was recirculated but it was changed in every experiment. Also a greater amount of water was sprayed per unit volume of air handled, than in commercial air washers.

A common question has been the effect of water purity of the air washer. This was an important factor, and one or two of you probably missed it in the paper. You will find on page 146 the importance of keeping the water fresh, that is changing the water in the air washer every day, or as, in our case, after every experiment.

Exposure in all experiments was 3½ hours as indicated on the slides. In most instances it was not necessary to continue the experiments that long, but we did, nevertheless. Equilibrium was established in one to three hours, depending on the air space per person and amount of outside air introduced to the room.

I fail to understand Mr. Haas' question. In our experiments, the minimum air space per occupant was 100 cu ft, and the maximum, 470 cu ft. The corresponding ventilation requirements were 25 and 7 cfm per person respectively. In other words, decreasing the air space per occupant to about one-fifth, increased the ventilation requirements three and a half times, not six times, as Mr. Haas' question would seem to imply. Perhaps I misunderstood his question.

In regard to Mr. Fineran's question, he probably overlooked the fact that the air-washer chart holds for a total air circulation of 30 cfm per person, as shown clearly on the chart (Fig. 8). Increasing the total air circulation through the washer beyond 30 cfm per person would increase the odor-absorbing capacity of the washer, and the washer curve in Fig. 8 would fall below the present curve parallel to it. It is reasonable to assume that at some point beyond 30 cfm the odor of water acquired by the air in passing through the washer might predominate and mask the body odor, thus setting a limit to the deodorizing action of the washer.

In reply to Mr. Christman's question, odors emanating from rugs and furnishings in a room may not be conspicuous in the presence of body odors, or may themselves mask the body odor depending on their relative strength. There is no doubt that a certain minimum amount of air is necessary to remove odors from furnishings in rooms occupied by few persons. In many instances natural infiltration is sufficient for this purpose.

The quantitation and qualitative relationships between air-borne bacteria in occupied rooms and amount of outdoor air supply is now being studied by Dr. Wells at the Harvard School of Public Health.

No. 1032

AIRFOIL FAN CHARACTERISTICS

By W. A. Rowe* (*MEMBER*), Detroit, Mich.

THE airplane propeller is an efficient device when used for the purpose of producing end thrust for which it is intended. It provides both high acceleration and high velocity in the air stream. These attributes render it less suitable for a ventilating fan, if a highly efficient and quiet fan is to result. In the latter instance it is desirable to move a large volume of air at as low a velocity as possible, with consequent low end thrust.

Another serious fault of the airplane propeller, when used as a fan, is its characteristic swirl at the tips of the blades. A typical example in the case of a small desk or circulating fan is shown in Fig. 1.[1] Fig. 2 represents typical air flow distribution through an airplane propeller. There is a distinct negative flow beyond the tips of the blades with a sharp reversal of the air stream as it returns back through the blades. When the same kind of blade is used in a fan the same condition is obtained. This blade design prevents the most effective part of the blade from doing its share of the work of discharging its quota of air through the orifice (Fig. 3). This factor may be eliminated by a radical change in the design of the blade tips, the orifice, or of both (Fig. 4).

The contraction in the area of the air stream passing through and beyond the orifice ring has the effect of increasing the effluent air velocity. In the case of a fan discharging to the atmosphere, all of the energy represented by the velocity head in the air stream is lost. Anything that can be done to lessen the contraction in the air stream will reduce the velocity pressure loss and increase the fan efficiency.

In the case of the airplane propeller the diameter of the vena contracta will amount to 85 per cent of the blade diameter. In a fan with modified blade tips, sufficient to eliminate the tip swirl, the minimum diameter of the air stream will be nearer 90 per cent, and with the addition of a carefully proportioned streamline orifice may reach a value of 95 per cent, which is approximately the limits to be obtained. The latter value corresponds to about 90 per cent of the area; the remaining 10 per cent represents the dead area at the center of the fan blades, where there is no measurable air velocity.

Attempts have been made to increase the effectiveness of the center third diameter of the blades, where the blade velocity is low, in an effort to improve

* Mechanical Engineer.

[1] Study of the Flow of Air with a Stroboscope, Harold E. Edgerton, *Mechanical Engineering*, April, 1935. Fig. 1 reproduced by permission.

Presented at 42nd Annual Meeting of the American Society of Heating and Ventilating Engineers, Chicago, Ill., January, 1936.

FIG. 1. FLOW OF AIR THROUGH TIP OF FAN

the fan efficiency. Unfortunately this is the region where the blade angle is already high, and any further increase in its angle of attack only serves to aggravate the disturbance to air flow at the back of the blade. A comparison of the continuity of air flow at low angles of attack as compared to the conditions which occur at larger angles is illustrated in Figs. 5 and 6.

One method of reducing the effluent air velocity is with the use of a discharge duct or collar. If the collar is made of equal diameter to the orifice and not less than one diameter in length, the air stream will expand beyond the point of greatest contraction and fill the duct. This reduces the air velocity and the velocity head, and increases the mechanical efficiency proportionately.

Results of comparative tests of a fan with and without a collar on the discharge is shown in Fig. 7. With the collar there is an increase in the efficiency at the free delivery capacity without the collar, viz. 6000 cfm, of 52

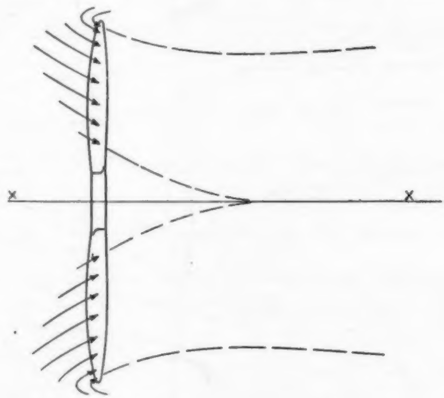

FIG. 2. AIR FLOW DISTRIBUTION THROUGH AIRPLANE PROPELLER

FIG. 1. FLOW OF AIR THROUGH TIP OF FAN

per cent, as compared with the same fan discharging to atmosphere without the collar. With this fan the diameter of the air stream at its point of greatest contraction is 90 per cent of the orifice ring diameter. The area ratio is 81 per cent, the velocity ratio 1.23, and the velocity pressure ratio 1.52. It will be noted that the last value is identical with the increase in the mechanical efficiency.

At other points of operation than at free delivery the increase in the efficiency will be found to be in proportion to total pressure ratios, as indicated in Table I.

The agreement of the results when the fan is credited with the actual air velocity it is producing with the results of the test with the collar indicate excellent recovery of velocity head in the high ratios of volume, and reducing toward no delivery.

If in the previous example a fan had been used giving a higher air stream coefficient, the decrease in velocity would have been less, and the possible improvement in the efficiency would have been less using the collar.

The collar also serves a second function by effectively eliminating the tip swirl of the fan. With the addition of the collar a situation results where the lower the fan efficiency, the greater the possible improvement. The Fan Test Code [2] provides that fan ratings made from tests with collar shall specify *For installation with ducts*. Generally the fan user is not aware of the fact that the ratings are totally unrepresentative of the performance of the fan when wall mounted and discharging to atmosphere. In all such cases ratings for both methods of application should be given.

A two blade airfoil revolving about its own axis is graphically represented in Fig. 8. As the blade revolves any element of an area at any radius r will,

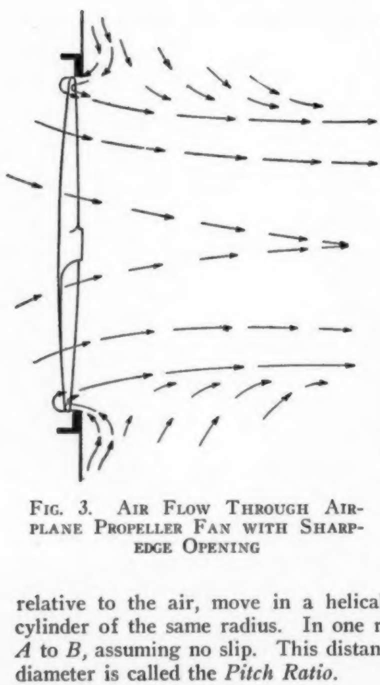

FIG. 3. AIR FLOW THROUGH AIRPLANE PROPELLER FAN WITH SHARP-EDGE OPENING

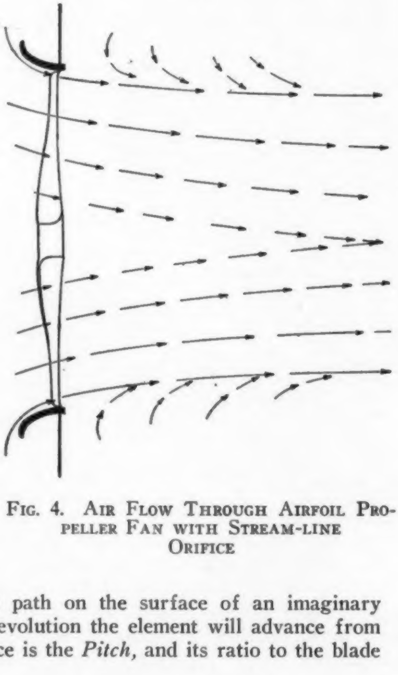

FIG. 4. AIR FLOW THROUGH AIRFOIL PROPELLER FAN WITH STREAM-LINE ORIFICE

relative to the air, move in a helical path on the surface of an imaginary cylinder of the same radius. In one revolution the element will advance from A to B, assuming no slip. This distance is the *Pitch,* and its ratio to the blade diameter is called the *Pitch Ratio.*

If the surface of the cylinder is assumed to be developed, the path of the element will be indicated by the diagonal, and the angle between the long side of the rectangle and the diagonal will be the same as the angle of the blade element referred to the plane of rotation. Then

$$Tan\ b = \frac{Pitch}{2\pi r.} \tag{1}$$

[2] A. S. H. V. E. Standard Test Code for Disc and Propeller Fans, Centrifugal Fans and Blowers, A. S. H. V. E. TRANSACTIONS, Vol. 29, 1923.

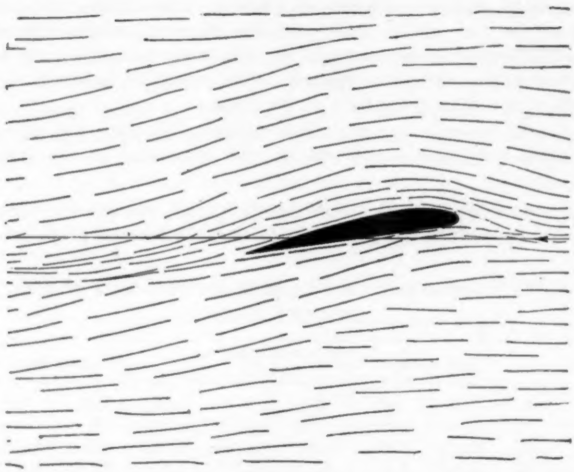

FIG. 5. FAN BLADE AIR FLOW WITH LOW ANGLE

Referring to Fig. 9 and plotting as abscissa the distance $\frac{Pitch}{2\pi}$ from the axis of the blade, any diagonal to that point from any element of area at any radius r will include an angle with the blade's vertical axis equal to the angle

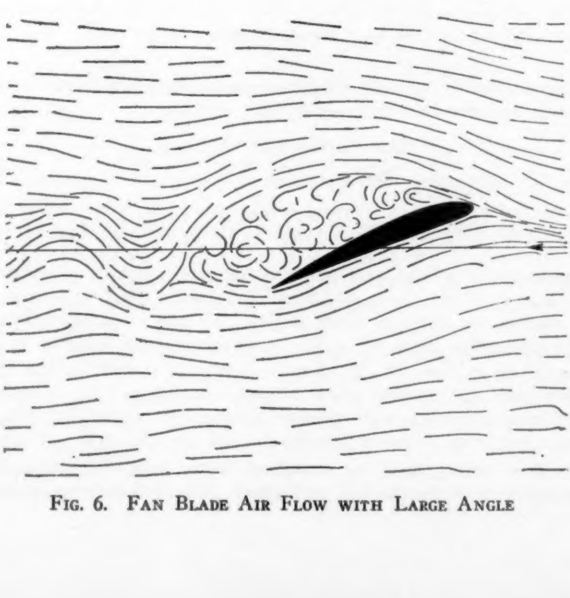

FIG. 6. FAN BLADE AIR FLOW WITH LARGE ANGLE

of the blade element with the plane of rotation. With blades of constant pitch the angle of the face of the blade with the plane of rotation continually increases as the center is approached. In that case, unless low tip angles are started, blade angles are soon reached, as the center is approached, too great for smooth stream flow at the back of the blade.

Where the blade angle is maintained constant, the pitch will be variable at different radii, being greatest at the tip and decreasing as the center is approached.

There is a third type in which both the blade angle and the pitch are variable, and it is this type which has proven to be suitable for fan purposes.

TABLE 1. COMPARATIVE TEST RESULTS OF AIRPLANE FAN WITH AND WITHOUT COLLAR [a]

Test without Collar							Test with Collar		
CFM	Vel.	V.P.	S.P.	T.P.	H.P.	M.E.	S.P.	T.P.	M.E.
6000	1407	0.120	0	0.120	0.255	0.445	0.062	0.182	0.675
5000	1170	0.083	0.092	0.175	0.251	0.55	0.135	0.218	0.685
4000	938	0.053	0.167	0.220	0.242	0.575	0.19	0.243	0.633
3000	705	0.030	0.223	0.253	0.23	0.520	0.23	0.26	0.533
When Velocity Is Increased 23 Per Cent and V.P. 52 Per Cent									
6000	1730	0.180	0	0.180	0.255	0.675			
5000	1440	0.125	0.092	0.217	0.251	0.683			
4000	1155	0.080	0.167	0.247	0.242	0.643			
3000	865	0.045	0.223	0.268	0.23	0.550			

[a] Air Weight = 0.0725 lb per cubic foot. Velocity Constant = 4072.

With any uniformly varying pitch the mean pitch will occur at two-thirds of the radius and can be expressed by the formula:

$$Mean\ Pitch = 2/3\,\pi\, D\ Tan\ b, \tag{2}$$

where b is the blade angle with the plane of rotation at ⅔ radius. Then

$$Pitch\ Ratio = \frac{2/3\,\pi\, D\ Tan\ b}{D} = 2/3\,\pi\ Tan\ b \tag{3}$$

Development of an efficient fan involves the determination of optimum values or proportions of blade form, and area distribution, thickness ratio and type of airfoil section for different blade angles, and rate of pitch variation. Also an efficient streamline form of orifice, with its position relative to the blade. The charts shown in Figs. 10, 11, 12 and 13 serve to compare effect of different number of blades and various pitch ratios from 0.5 to 1.0. It will be noted that the volume increases with greater number of blades, but not in direct proportion. For free delivery condition, and for the type of airfoil blade covered by this investigation, an empirical factor of $\sqrt[4]{B/2}$ is approximately correct, in which B is the number of blades. This will correspond to approxi-

mately 20 per cent more capacity with four blades as compared with two, and 30 per cent for six blades as with two.

The static pressure in the region of restricted deliveries will increase approximately in direct proportion to the number of blades for equal pitch ratios.

The horsepower curves are fairly flat for low pitch ratios, increasing toward restricted deliveries as the pitch ratio is increased, and slightly also as the number of blades is increased. This design characteristic requires no larger capacity motors for operation against resistances than for free delivery, except for six blade fans at the higher pitch ratios. In the latter case a moderate increase in the motor size of 25 per cent will suffice, except at low ratios of volume.

The mechanical efficiencies are quite similar for different numbers of blades

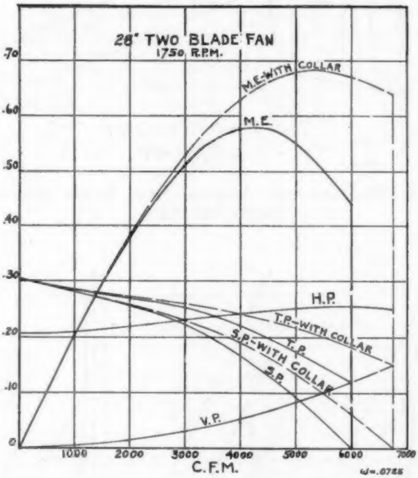

Fig. 7. Airplane Fan Performance Curves with and without Discharge Collar

at the same pitch ratios, the increase for four blades as compared to two being approximately two per cent, and a similar increase for six blades as compared to four. This difference in efficiency is somewhat at variance with published efficiency data on airplane propellers, which is based on thrust. In the case of ventilating fans additional blades serve to increase the coefficient of area of the effluent air stream. Due to these design characteristics it is possible to realize more favorable efficiencies at and near to free delivery, where most of these fans operate. The mechanical efficiencies which result from the prescribed method of computation, using the full area of the orifice for determining the average velocity and the velocity pressure, result in apparently higher relative efficiencies for restricted deliveries than for free delivery. Where the actual velocities and calculations are determined, it will be found that the efficiency curves tend to flatten out at the peak, giving the same rela-

tive advantages for increase in number of blades at correspondingly restricted conditions of operation as for free delivery. Computations of this kind raise the efficiency the greatest amount at free delivery, where the total head is all velocity pressure, whereas at all other points the velocity head becomes a relatively smaller proportion of the total pressure as the static pressure increases. (See Table 1.)

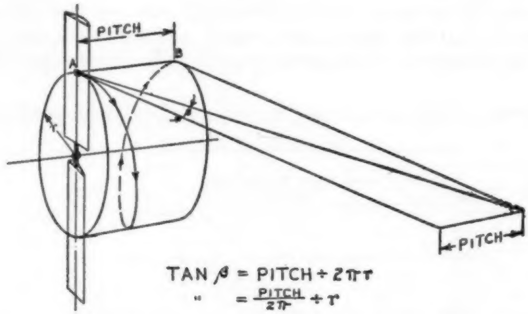

FIG. 8. DIAGRAM OF AIRFOIL FAN BLADE REVOLVING ABOUT ITS AXIS

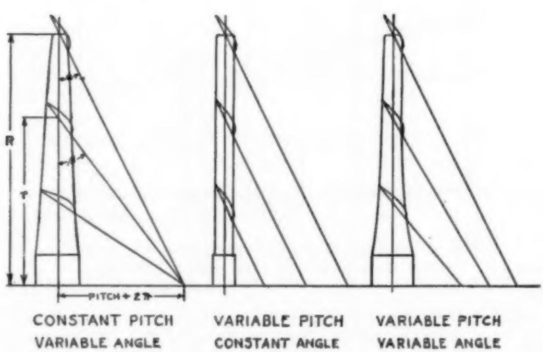

FIG. 9. AIRFOIL FAN BLADE PITCH AND ANGLE RELATIONSHIPS

The static efficiencies, while not shown on the curves, will obviously be greater for low than for high pitch ratios.

The decreased efficiencies at higher pitch ratios would appear to reflect the importance of avoiding the disturbance to stream flow which accompanies higher angles of attack. The best pitch ratio for free delivery will be found to be 0.75, corresponding to a mean blade angle at ⅔ radius of 20 deg.

In any propeller operating in air, as in water, there will be a certain slip.

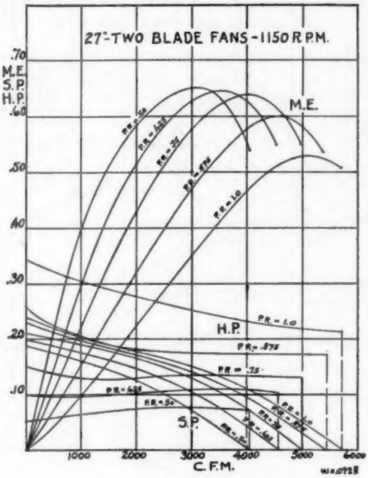

FIG. 10. PERFORMANCE CURVES OF 27 IN. TWO BLADE FANS, 1150 RPM.

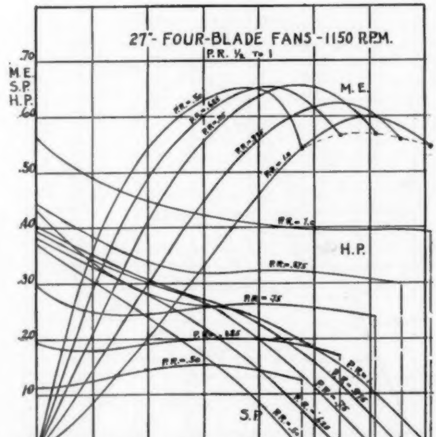

FIG. 11. PERFORMANCE CURVES OF 27 IN. FOUR BLADE FANS, 1150 RPM

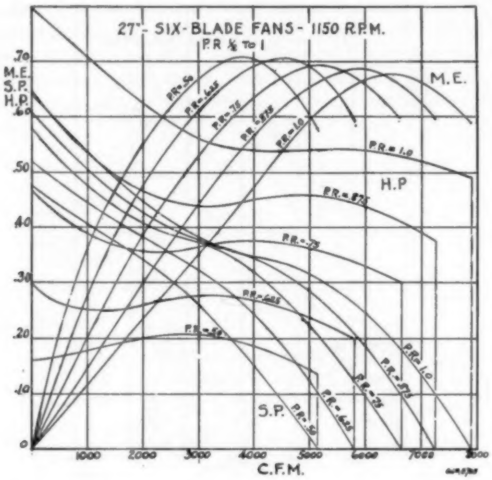

FIG. 12. PERFORMANCE CURVES OF 27 IN. SIX BLADE FANS, 1150 RPM

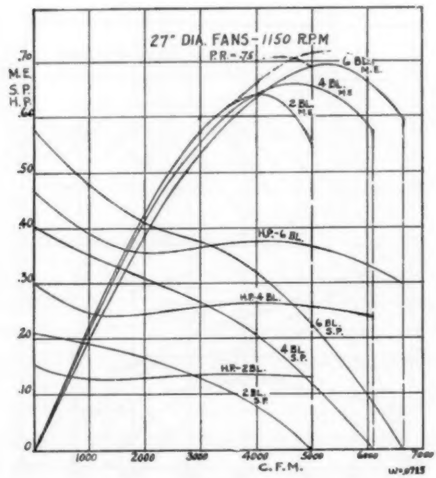

FIG. 13. COMPARISON CURVES OF TWO, FOUR AND SIX BLADE FANS, PITCH RATIO 0.75 AND MEAN BLADE ANGLE 20 DEG

The geometrical pitch minus the slip is called the true pitch. This can be expressed thus:

$$True\ Pitch = Geometrical\ Pitch\ (1\text{-}Slip) \tag{4}$$

The true pitch multiplied by the RPM gives the velocity or the distance the particle moves in one minute. The volume of air handled by the fan at free delivery may be expressed as follows:

$$\text{CFM} = 2/3\,\pi\ D\ N\ tan\ b\ A\ (\text{1-Slip})\ \text{where}\ A = \text{area of orifice} \tag{5}$$

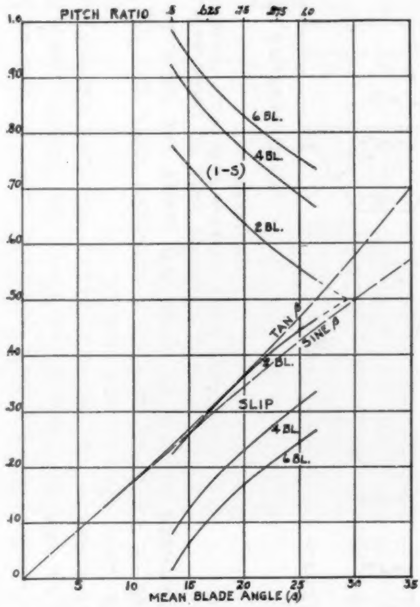

FIG. 14. COMPARATIVE CURVES OF SLIP AND PITCH RATIOS FOR TWO, FOUR AND SIX BLADE FANS

If the value $\dfrac{\pi D^2}{4}$ is substituted for A the formula is:

$$\text{CFM} = 1.645\ D^3\ N\ Tan\ b\ (1\text{-}S) \tag{6}$$

If test results values of calculated slip are plotted against pitch ratios, or mean blade angles, or both, a series of curves will be obtained for two, four, and six blades as shown in Fig. 14.

In the case of the two blade fans the slip will be found equal to the tangent of the mean blade angle for optimum pitch ratios, and equal to the sine of the angle at pitch ratios below and above the optimum pitch ratios. These

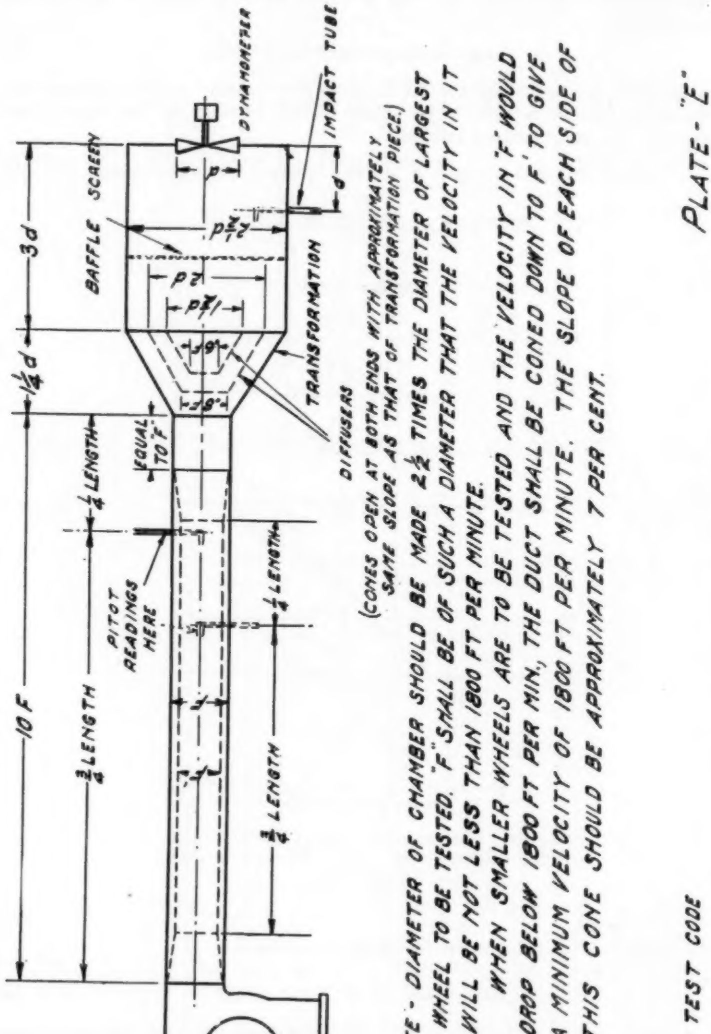

FIG. 15. DIAGRAM OF STANDARD TEST SET-UP FOR DISC AND PROPELLER FANS

values may then be substituted in formula (6), and if greater accuracy is desired the following values of S may be used:

Mean Blade Angles 12 to 17 deg = *Sine b*
Mean Blade Angles 17 to 22 deg = *Tan b*
Mean Blade Angles 22 to 27 deg = (*Sine b* + *Tan b*) ÷ 2
Mean Blade Angles 27 to 32 deg = *Sine b*

For four and six blade fans the value of the slip may be determined from the chart, or the empirical value $\sqrt[4]{\frac{B}{2}}$ as previously mentioned may be used. The formula is then applicable to any number of blades, and to all pitch ratios, or mean blade angles, covered by these data.

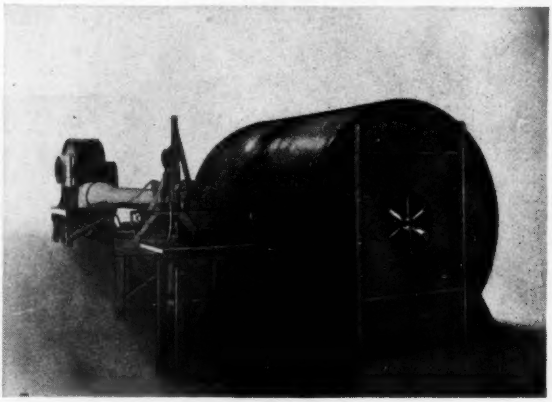

FIG. 16. GENERAL VIEW OF FAN TEST APPARATUS

(For ease in converting values of pitch ratio to corresponding mean blade angles reference is again made to the formula *Pitch Ratio* = $\frac{2}{3}\pi$ *Tan b*.)

All data presented herein are from tests made in accordance with the provisions of the Standard Test Code for Centrifugal and Propeller Fans. In Fig. 15 is shown a diagram of Plate E from the Test Code, and an illustration of the actual test set-up is shown in Fig. 16.

These performance data do not apply to all airplane or airfoil propeller fans, but only to the particular design herein presented, and having a blade of substantially constant width, except for the widening projection at the tip, as illustrated in Fig. 17.

The airfoil blades were cast of aluminum alloy from metal patterns and accurately finished to gages for correct airfoil section. They were then buffed and polished, care being taken to insure exact wheel balance without the use of any balance weights. Blade angles are established within $\frac{1}{5}$ of a degree at five stations equidistant from one-third radius to the tip.

The use of more favorable blade angles for efficient operation as compared to the more common practice with other types requires somewhat higher speeds for airfoil fans. However, the airfoil itself may be run at slightly higher speeds for equal quietness as compared either with thin plates or high pitch ratios. Referring to Figs. 5 and 6 it will be noted that each revolving blade must pass through the wake of the preceding blade. It is therefore important for quiet operation that the disturbance at the back of the blades be kept as small as possible.

With two blade fans at ordinary speeds there may be a noticeable flutter, due to the low frequency of the vibration. When an increased number of blades are used, this disappears, as the ear does not readily separate higher frequency sounds from all the several variable factors involved in the total

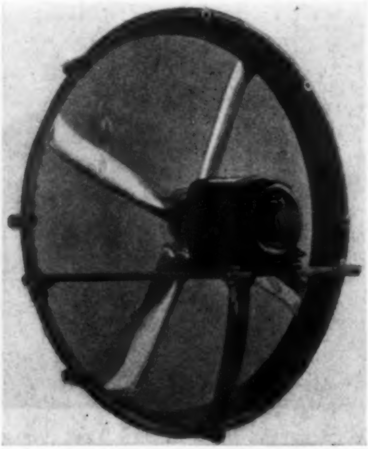

FIG. 17. FOUR BLADE AIRFOIL FAN

noise volume. Within the limits covered by these tests it was found that better results may be accomplished with a greater number of blades, even though a slightly higher decibel rating may result. A four blade airfoil fan is shown in Fig. 17 and a six blade direct connected fan unit in Fig. 18.

The same judgment is necessary in applying these fans as with any other type where quiet operation is required. Generally low blade velocities tend towards quietness, and high velocities towards noise. Quiet motors with resilient mountings and sleeve bearings are used for best results. When capacitor motors are used, additional speeds are available with a slight additional expense as compared to single speed motors.

In many of the industrial applications, where the noise level of the environment is relatively higher, usual standard motor speeds, windings, and ball bearings may be used.

CONCLUSIONS

The ordinary airplane propeller is inherently unsuited for use as an efficient ventilating fan without modification of either its form of blade tip, or the orifice ring, or both.

The contraction of the air stream increases the velocity head loss. Increasing the coefficient of area of the air stream raises the mechanical efficiency.

Streamline orifices, more effective forms of blade tips, and a greater number of blades all tend to increase the efficiency.

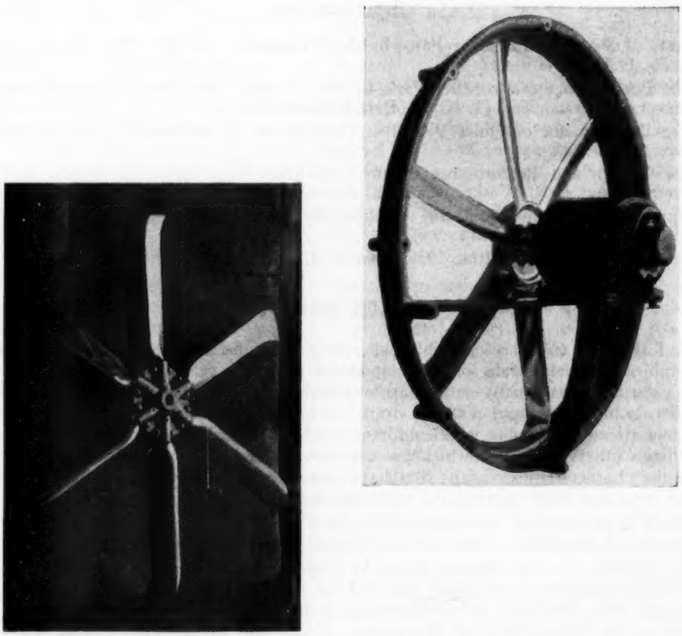

FIG. 18. SIX BLADE AIRFOIL FANS

The use of a short discharge duct eliminates swirl at blade tips, recovers some velocity head, and increases the capacity and the mechanical efficiency.

Variable blade pitch is better than constant pitch and moderate pitch ratios of 0.75 are the optimum.

An increase in the number of blades increases the capacity as the fourth root of the number ratio, the static pressure more nearly in direct proportion to the number, and the mechanical efficiency slightly.

With airfoil blades high mechanical efficiencies may be realized and especially in the region of free delivery where the majority of these fans operate.

Two blade fans are not as quiet as four or six blades because of the low frequency vibration causing a noticeable flutter.

Fans with airfoil blades have non-overloading horse power characteristics except with multi-blades having pitch ratios above the optimum.

Airfoil fans conform to *Fan Laws,* in relation to capacity, pressure, and horse power when the speed, air density, or size is changed. As the great majority of these fans operate at, or near, free delivery, and are frequently only so rated, these *Fan Laws* are very helpful in calculations concerned with the selection and application of fans.

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DISCUSSION

A. I. BROWN (WRITTEN): This paper is a welcome addition to the literature on the subject of airfoil fans or more specifically on the subject of airplane propeller fans. An airfoil is usually defined as "any body so shaped that the cross force, called the lift, is high compared with the drag." The word *airfoil* has such a broad meaning that it is questionable whether airfoil fans can be said to have any certain characteristics other than those which are common to fans of all types. The paper deals with the characteristics of fans that have come to be widely known as airplane propeller fans and more particularly with one design of an airplane propeller fan.

Much of the material presented in the paper is not new, but nevertheless, I believe, is worthy of repetition. The various means for increasing the efficiency of the airplane propeller as a ventilating device by proper streamlining of the orifice ring and by effective design of the blade tips and by the addition of a short discharge duct have all been described and discussed in a bulletin [3] to which reference has been made in the bibliography of this paper. Since the publication of that bulletin a number of manufacturers of airplane propeller fans have modified their designs along the lines suggested with resulting improvement in performance, as in the case described in Mr. Rowe's paper.

It is now some 14 years or more since airplane propeller fans first appeared on the market. The first fan of this sort, like many which followed, had no apparent means for preventing the contraction of the stream of air and the consequent backflow at the tips of the blades. The performance was accordingly poor and the manufacturer soon discontinued the manufacture. It is unfortunate that he did not, instead, discover the reasons for the poor performance and correct the faults.

Mr. Rowe's paper and the papers to which reference has been given show conclusively that when the faults of the airplane propeller as a ventilating device have been properly corrected it is transformed from an inefficient into an efficient fan.

[3] Ohio State University *Engineering Experiment Station Bulletin,* May, 1933.

The paper shows very clearly the influences of the pitch ratio and the number of blades upon the shape of the power curves. It is probable that these influences produce similar effects in the characteristics of all airfoil fans; but it should be noted that the shapes of the power curves shown in Figs. 10, 11, and 12 apply specifically to the fan tested, or to fans with blades of a similar shape, and not to all airfoil fans or even to all airplane propeller fans. The shape of the power curve depends not only upon the pitch ratio and the number of blades, but upon a number of other variables which include the planform of the blades, the shape of the airfoil sections, and the variation in pitch, thickness, and camber along the length of the blade. In fact, a comparison of the shapes of the power curves shown in this paper and those shown in THE GUIDE and in Professor Marks' paper and the writer's bulletin, show marked differences. The differences appear to be due not so much to the pitch ratio as to the other variables for which no data appear in this paper.

One of the outstanding characteristics of airplane propeller fans has been the fortunate non-overloading feature. At no point in the range of capacity has the power requirement exceeded that for the free-discharge capacity or the capacity corresponding to the point of highest mechanical efficiency. The only exception to this performance has been in the case of a few multiblade fans with high pitch ratio, but even here the increase in power required for no-discharge has been so slight as to be of no consequence. This desirable feature is lacking in some of the power curves of Figs. 10, 11, and 12, especially in the case of the four and six blade fans, where in the worst case the power consumption for no-discharge is about 63 per cent greater than for free discharge, or almost 50 per cent greater than for the point of highest mechanical efficiency.

Marks and Weske showed in their paper[4] the power curve for a three-blade airfoil fan which also lacked the non-overloading feature, but the blades of this fan were very different in shape from airplane propeller blades, being widest at the tips. Even though the fan developed a maximum efficiency of 80 per cent, its performance was considered unsatisfactory because of the increase in power which accompanied an increase in resistance. An attempt has been made to overcome this defect, and the results are reported in a more recent paper.[5] In the redesign the most conspicuous change was in the planform of the blades, the width at the tips being greatly reduced. Even though the number of blades was increased from three to eight, with little change in the pitch ratio, the power curve was flattened to the extent that the power required for no-discharge was reduced from 225 per cent to 116 per cent of the power required at the point of maximum efficiency. This improvement in power characteristic was gained at a sacrifice of only 2.5 per cent in maximum fan efficiency.

Professor Marks's work adds emphasis to the desirability of the non-overloading feature which has been generally characteristic of airplane propeller fans, and also suggests that by variations in the design of blades it is possible to maintain the non-overloading characteristic along with high efficiency even when the pitch ratio is high.

Mr. Rowe's paper is valuable in showing that when the characteristic curves have been established for a fan with a certain pitch ratio and number of blades, the curves for another fan of similar design but with a different pitch ratio or with a different number of blades can be readily predicted.

W. A. ROWE: The Marks and Weske fans Professor Brown discussed were the pressure types and not the fans which I have referred to in this paper. They discharged into a Venturi with possibly diffusion vanes and other methods for conversion of pressure, and were fans with large blade area ratio. Such fans are used more or

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[4] The Design and Performance of an Axial-Flow Fan, by L. S. Marks and J. R. Weske, A. S. M. E. TRANSACTIONS, Vol. 56, 1934.

[5] The Design of a High Pressure Axial-Flow Fan, by L. S. Marks and Thomas Flint, A. S. M. E. TRANSACTIONS, Vol. 57, 1935.

less as under-grate blowers, for which service their high speed characteristics lend themselves favorably to direct turbine drive.

The airfoil ventilating types of fans covered by this paper are used for operation near free delivery. Any discussion as to their horsepower requirements at, or near, no discharge is purely academic. They are not so applied.

No. 1033

ROOM SURFACE TEMPERATURE OF GLASS IN WINDOWS

By J. E. Emswiler,* Ann Arbor, Mich., and W. C. Randall,** Detroit, Mich. (*MEMBERS*)

The results presented in this paper were obtained in connection with a project sponsored by the Technical Committee of the Metal Window Institute and conducted at the University of Michigan in the Department of Engineering Research. Special acknowledgment is due the following members of the Institute: Detroit Steel Products Co., Truscon Steel Co., William Bayley Co., Hope's Windows, Inc., Concrete Engineering Co., Crittal Manufacturing Co., and Mesker Brothers' Iron Co., who contributed to this research program.

THE room surface temperature of glass in windows is determined by a percentage of the total difference in temperature between inside and outside air. Actual test results, almost without exception, indicate that this percentage is a constant factor for any particular kind of window and wind condition, and for all temperatures of outside air. The use of factors representing the various percentages or ratios will allow the solution of problems concerning the depositions of moisture on single, double glazed windows or double windows, as well as other problems involved in the use of windows for the modern air-conditioned installations.

The increasing use of air-conditioning with relatively high humidities has made imperative the solution of the problem of the factors involved in the deposition of moisture on the room surface of glass in windows. Engineers and manufacturers of windows are frequently requested to make recommendations regarding the elimination or control of this excessive deposition.

Experimental Determination of Factors for Room Surface Temperatures

The apparatus for the tests consisted of a *cold box,* in which low temperatures could be produced by dry ice (solid carbon dioxide) placed in bunkers; and a *warm box,* in which room temperature could be maintained by electric

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FIG. 1. WINDOW SIDE OF COLD BOX SHOWING LOCATION OF THERMOCOUPLES

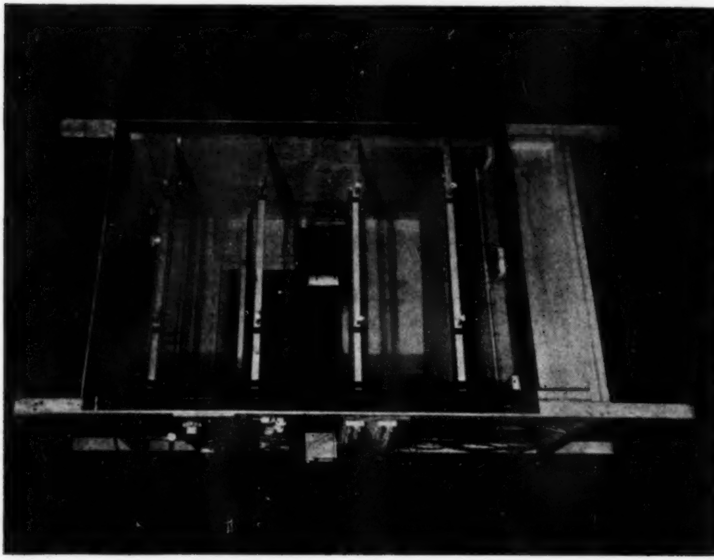

FIG. 2. VIEW OF WARM BOX SHOWING THERMOCOUPLES AND HUMIDITY RECORDER

heaters. An opening was provided for a window with dimensions of approximately 40 in. wide by 54 in. high and was located in the wall separating the warm and cold boxes. Temperatures were obtained by thermocouples, as explained in the following description of the apparatus.

Humidity of the air in the warm box was measured by wet- and dry-bulb thermometers. Determining dew point temperature from air temperature and humidity, and observing when moisture first appeared on a window under test, gave an independent check of observations from thermocouples on the windows. A similar check was obtained when frost first appeared.

Moisture was deposited upon the window surfaces in many of the single window tests, and ice was often present with low cold box temperatures. Their presence on the window seemed to produce little or no effect upon temperature relations, in other words, the temperature drop did not seem to be affected by moisture or frost.

Each window was tested at three different cold (outside) air temperatures, about 35 F, 15 F, and —5 F in the cold box with and without wind, and with warm air temperature in the warm box kept at room temperature.

APPARATUS AND METHOD OF TESTING

The test apparatus consisted of a *cold box* to simulate outside conditions and a *warm box* to simulate room conditions, with the window under test mounted in a frame between them, and the necessary thermocouples and thermometers to measure the desired temperatures.

The cold box was 8 ft 6 in. high by 5 ft square, built of 1 in. matched lumber on 2 in. x 4 in. framing, and covered with 1½ in. of insulating board. The window under test was installed in a 54½ in. x 40 in. opening in the front of the box. Cooling was accomplished by the use of carbon dioxide ice placed in galvanized sheet metal bunkers inside the box at the top and back. The carbon dioxide gas evolved was piped off to a manifold on the side of the box and discharged out-of-doors. Cold box temperatures were controlled by addition or removal of ice from the bunkers.

Wind on the outside surface of the window was produced by a motor driven blower inside the cold box, with the outlet connected to two vertical headers having holes drilled in them to direct the streams of air toward the window surface at an angle of approximately 45 deg. Iron constantan thermocouples, guarded against radiation, were located at four levels on the cold side of the window, spaced 6 in. from the glass surface, as shown in Fig. 1. All thermocouples were made of No. 28 wire.

White curtains were hung in the cold box to prevent direct radiation to the extremely cold bunker surfaces. A gasket of rubber tubing fastened to the frame around the window opening sealed the joint where the two boxes were clamped together.

The warm box was 89 in. high, 44 in. wide, and 32 in. deep, built of ⅜ in. plywood on 2 in. x 2 in. framing, covered with ½ in. of insulating board. A 54½ in. by 40 in. opening in the front fit the window opening in the cold box. A 2 ft square observation window was mounted in back of the box.

Heat was supplied by electric strip heaters in the bottom of the box, guarded against direct radiation to the windows and box walls. A variable resistance

outside the box was used to control the electrical input to maintain constant temperatures in the warm box.

Relative humidity was determined from readings of wet- and dry-bulb thermometers on a shelf in the warm box visible through the observation window. A small blower was used to circulate air across the thermometer bulbs.

Thermocouples were located 6 in. from the room surface of the glass at four zones opposite the cold box thermocouples as shown in Fig. 2. Glass temperatures were obtained from thermocouples flattened and secured to the glass surface with thin cement.

FIG. 3. WARM BOX AND COLD BOX PLACED TOGETHER FOR TEST

A compensated potentiometer indicator was used with a calibration chart to obtain the temperatures at the various thermocouple locations. Double pole knife switches as shown in Fig. 3 were used to select the desired thermocouples for temperature measurements.

Test runs were of two to three hours duration at constant temperature conditions with readings recorded every 15 min. Two and a half to three hours time was allowed before each test to reach stable conditions.

THE FACTOR R FOR ROOM SURFACE TEMPERATURE OF GLASS IN WINDOWS

The values of this factor or ratio R were obtained by substituting test results in the following formula:

$$R = \frac{t_r - t_g}{t_r - t_o} = \frac{\text{room temperature—temperature room side of glass}}{\text{room temperature—temperature of outside air}} \tag{1}$$

Sketches A and B of Fig. 4 indicate the approximate places at which thermocouples were placed on the glass.

Typical examples of test observations and results from two of the windows are shown in Table 1.

VALUE OF R AS DETERMINED FROM TESTS

The charts of Fig. 5 (with wind) and Fig. 6 (without wind), present the values of R, as found from the tests, in a graphical form. In explanation, the

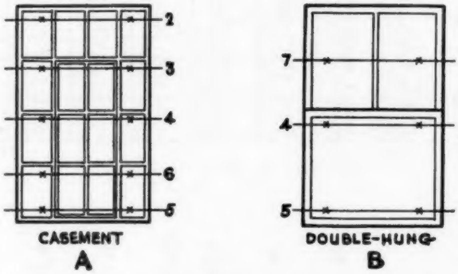

FIG. 4. SKETCHES SHOWING LOCATION OF THERMOCOUPLES ON GLASS SURFACE ON ROOM SIDE OF WINDOW

scale at the top of the charts, extending from 0 to 100, may be considered as *per cent*. The total length of each bar is 100, and represents the total temperature difference ($t_r - t_o$) between inside and outside temperature through a window, from warm to cold air. The length of the cross-hatched portion of each bar represents the drop in temperature from warm air to room surface of glass ($t_r - t_g$). Since the total length of a bar is 100 per cent, the length of the cross-hatched part also represents the value of R, that is the ratio $\frac{t_r - t_g}{t_r - t_o}$ in per cent, for that particular window.

Inspection of the charts of Figs. 5 and 6 shows that the values of R fall into two rather well-defined groups, that is single windows and double windows, the latter including double glazed windows, all of which are summarized in Table 2.

TYPICAL EXAMPLES OF THE USE OF FACTOR R

Formula 1 shows how the factor R is derived from test data (as illustrated in Table 2) for different types of windows. The equation or the formulae may

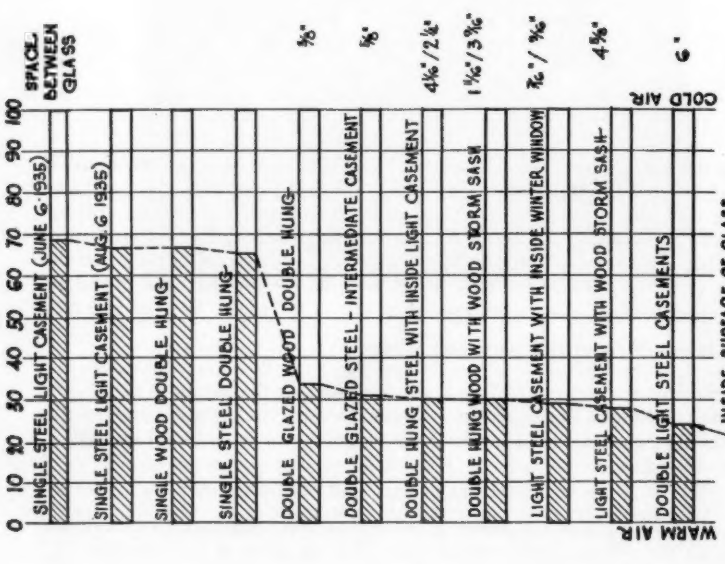

Fig. 5. Values of R With Wind

$$R = \frac{t_r - t_g}{t_r - t_o} = \frac{\text{Room Temperature — Temperature Room Side of Glass}}{\text{Room Temperature — Temperature of Outside Air}}$$

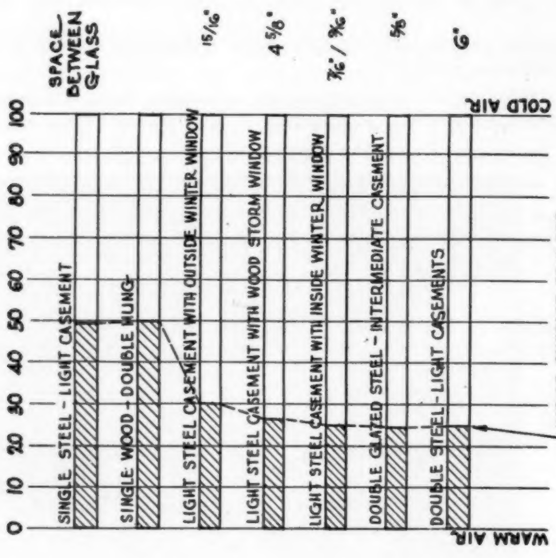

Fig. 6. Values of R Without Wind

$$R = \frac{t_r - t_g}{t_r - t_o} = \frac{\text{Room Temperature — Temperature Room Side of Glass}}{\text{Room Temperature — Temperature of Outside Air}}$$

Table 1. Temperature Results of Single Double Hung Wood and Light Steel Casement Windows

Windows	Single Double Hung Wood			Single Light Steel Casement		
Test No.	42	43	44	57	58	59
Air Temperature (Average)						
t_r	75.9	74.3	71.7	73.9	73.9	71.4
t_o	36.4	17.3	− 2.3	36.9	18.6	− 0.5
Glass Temperature (Average)						
t_2	...	...	...	49.5	38.1	24.2
t_3	...	...	...	47.8	34.6	20.3
t_4	51.8	39.2	26.8	49.2	36.5	22.8
t_5	49.8	36.4	23.9	49.6	36.4	23.8
t_6	...	...	...	50.7	37.2	24.9
t_7	47.0	32.3	17.9	...	...	...
Temperature Drops						
$t_r - t_o$	39.5	57.0	74.0	37.0	55.3	71.9
$t_r - t_2$	...	...	...	24.4	35.8	47.2
$t_r - t_3$	...	...	...	26.1	39.3	51.1
$t_r - t_4$	24.1	35.1	44.9	24.7	37.4	48.6
$t_r - t_5$	26.1	37.9	47.8	24.3	37.5	47.6
$t_r - t_6$	...	...	...	23.2	36.7	46.5
$t_r - t_7$	28.9	42.0	53.8	...	...	...
Ratio of Temperature Drops (Factor R in per cent)						
$(t_r - t_2) \div (t_r - t_o)$	...	...	...	66	65	66
$(t_r - t_3) \div (t_r - t_o)$	...	...	...	71	71	71
$(t_r - t_4) \div (t_r - t_o)$	61.0	61.5	61.0	67	68	67
$(t_r - t_5) \div (t_r - t_o)$	66.0	66.5	65.0	65	68	66
$(t_r - t_6) \div (t_r - t_o)$	...	...	...	63	66	65
$(t_r - t_7) \div (t_r - t_o)$	73.0	74.0	73.0	...	...	...
Average Factor R		66.8			67.0	

be rewritten for convenient use in solving a particular unknown value in a definite problem. For example, to determine outside temperature,

$$t_o = t_r - \frac{t_r - t_g}{R} \tag{2}$$

Table 2. Values of R for Glass for General Window Types

General Type of Window	Wind Condition	R for Glass in Per cent (Approx.)
Single Windows	With Wind	67
Double Windows	With Wind	30
Single Windows	Still Air	50
Double Windows	Still Air	25

The use of the factors will be illustrated by the following examples:

(*a*) At what outside temperature, with wind, will moisture begin to appear on the glass of a single window when the inside temperature is 70 F, and the relative humidity is 30 per cent?

From Fig. 5 (or Table 2) it is found that, for single windows, with wind, R is about 67 per cent. From a psychrometric chart, or by the use of steam tables, the dew point for 70 F and 30 per cent relative humidity is found to be 37 F, which would be the glass temperature at which moisture would begin to appear. Substituting these known values in equation (2), t_o, or outside air temperature, is found to be approximately 21 F.

(*b*) At what outside temperature will frost appear on the window of example (*a*) for the condition stated?

Condensation forming at a glass temperature of 37 F will be in the form of moisture. However, if the glass temperature is 32 F, or less, this will be in the form of frost. Using 32 F as t_g in formula (2) the outside temperature is found to be about 13 F.

(*c*) With a double steel window and with inside temperature 70 F and a relative humidity of 45 per cent, how low may the outside temperature go before moisture will appear on the glass, with wind?

The chart of Fig. 5 gives R for this window to be about 25 per cent and the dew point for the stated conditions is found from a psychrometric chart to be 47.5 F. These known values substituted in equation 2 give an outside temperature of —20 F. If the average factor of 30 per cent for double windows is used as given in Table 2, an outside temperature of —5 F is obtained.

TABLE 3. OUTSIDE TEMPERATURE AT WHICH DEPOSITION WILL OCCUR FOR 70 F INSIDE TEMPERATURE, WITH WIND

RELATIVE HUMIDITY IN PER CENT	F FOR DEW POINT OR t_g	F FOR SINGLE WINDOWS $R = 0.67$	F FOR DOUBLE WINDOWS $R = 0.30$
25	32.5	14.0	− 55.0
30	37.5	21.5	− 38.5
35	41.0	27.0	− 26.5
40	44.5	32.0	− 15.0
45	47.5	36.5	− 5.0
50	50.5	41.0	+ 5.0
55	53.5	45.5	+ 15.0

These examples indicate the practical application of the information derived from these experiments. It is possible to tabulate the results for problems of a similar nature, as shown in Table 3.

The conclusions that may be drawn from such a tabulation would be:

(1) For single windows and 30 per cent relative humidity, there will be deposition on the glass with an outside temperature of about 22 F. For a change in relative humidity of 1 per cent, the outside temperature at which moisture will form changes about 1 F.

(2) For double windows the relative humidity apparently must be more than 45 per cent to cause deposition when the outside temperature is 0 F. An increase in the relative humidity increases the outside temperature at which moisture will form by about 2 F for each 1 per cent increase in relative humidity.

(3) A further examination of Figs. 5 and 6 showing actual test results indicates that there is practically no difference in the room surface temperature of single glass in steel and in wood windows. There appears to be some difference in the cases of double glass although this may be accounted for by the difference in the air space between the panes of glass.

(4) A high degree of precision was not sought in these tests. Greater accuracy might have been attained by tests of longer duration, and with perhaps longer transition periods between tests. It is confidently believed, however, that the results obtained are well within the degree of precision warranted by the application of the data; and by dispensing with extreme refinement in the testing, it was possible to cover a much wider range of window types than could otherwise have been done with the time and funds available.

CHECK WITH GUIDE DATA

The value of the ratio R depends upon the ratio of surface coefficient for the glass. Let c = the ratio of the surface coefficient for wind (f_o) to that for still air (f_i). Then $c = f_o/f_i$. From equation in THE A.S.H.V.E. GUIDE 1935, page 93, and statements on pages 116 and 118, it can be shown that $R = \frac{c}{c+1}$ for single glass and $R = \frac{c}{3c+1}$ for double glass, when the resistance of the glass itself to heat transfer is considered to be zero.

Using the values $f_i = 1.50$, and $f_o = 4.50$, given in THE GUIDE previous to 1933, $c = f_o/f_i = 3$ and R, for single glass, $= 0.75$, and for double glass 0.30. It can also be readily shown that the overall coefficient of transmission, U is equal to $R \times f_i$, hence, for single windows, $U = 0.75 \times 1.50 = 1.125$ or 1.13, and for double windows, $U = 0.30 \times 1.50 = 0.45$, which are the values given in THE GUIDE 1935, page 113.

Later values of $f_i = 1.65$ and $f_o = 6.00$ are given in THE GUIDE 1935, page 98, and using these, R for single glass, is 0.78 and for double glass, 0.305. When the overall coefficient is computed, using $f_i = 1.65$ and $f_o = 6.00$, it is found that $U = 1.29$ for single glass, and $U = 0.50$ for double. Thus there is a discrepancy between the values of U and those of surface coefficient as given in THE GUIDE.

The value of $R = 0.67$ for single glass, based on test results as given in this paper, is less than $R = 0.75$ and 0.78 resulting from calculations based on surface coefficient values given in THE GUIDE. Possibly the wind effect upon the outside glass surface in these tests was less than that previously determined as produced by a 15 mph wind, which accounts for the smaller value of R. For double glass, the whole effect of the wind is minimized to the point where the test results and calculated values are closer.

The larger the value of R used, the higher will be the calculated temperature of outside air to cause deposition of moisture. Thus in Table 2, if R is taken at 0.75 instead of 0.67, the outside temperature to cause deposition on single windows, with 25 per cent relative humidity is 20 F; with 45 per cent, 40 F; and with 55 per cent, 48 F.

If the test value of R is correct, then the surface coefficients as now shown are incorrect. A revised value of U could be determined based on revised

values for surface coefficients if discrepancies of the magnitude indicated from these tests would warrant such a revision.

A point of interest is that the value of R of 0.67 for wind is reduced to 0.50 for no wind and for double glass from 0.30 to 0.25. This would seem to indicate that the actual heat loss would be reduced 25 per cent for single windows and 15 per cent for double windows for those not exposed to the wind.

Acknowledgment

The authors wish to acknowledge the work of L. V. Beukema, who conducted the laboratory work, also the cooperation of R. C. Montgomery, R. L. Carlson, R. L. Clingerman, E. W. Conover, and R. H. Sartor.

DISCUSSION

John James (Written): The collection of condensation between double glazed windows has presented difficult problems in many localities where double windows are permanently fixed in place. Due to the breathing in of the air at various temperature variations there has frequently occurred a collection of moisture on the interior surface of the two panes of glass. The problem that has arisen was how this objectionable feature could be successfully eliminated.

There has been some work attempted in vacuumizing the space between the two glasses. However, when the proportions of glass area are increased considerably the problem of preventing breakage of the glass is an important consideration. Also there has been the thought of decalorizers placed between the two windows, but, of course, over a period of time they collect all the moisture they are capable of absorbing and again the same problem arises.

Perhaps Mr. Randall can discuss this condensation problem in connection with this paper and give us his comments on actual experiences in the field.

W. C. Randall (Written): Generally speaking, I would recommend against the use of double glazing when installed in the manner which Mr. James has indicated. The objections are that it would be extremely difficult to keep the panes of glass clean, particularly the concealed or inside face to face surfaces, there will be breathing to the spaces between the lights of the inside humid air and, unfortunately, at times, condensation with no opportunity for the water to escape. The tendency would be for the space between the panes to become filled with water and dirt. If the inside or room conditions are such that the air is not high in humidity, this condition is not so acute as when high inside humidities are involved.

While I have never measured relative humidity in a railroad car, I assume that the relative humidity would be low, therefore, double glazing in a railroad car would involve less trouble such as you have given than for a typical air conditioned building.

Double glazing done with the greatest of care, such as the method used by Thermopane, minimizes the above conditions to a reasonable degree but trouble is not unusual. Many have tried, but none have accomplished, so far as I know, the general result of a vacuum between the lights of glass due to the tendency for the glass to collapse.

F. B. Rowley: The question of surface temperature is of particular importance when it comes to the problem of carrying high humidities in cold weather. As the author states, these surface temperatures may be calculated by subtracting from the room temperature a percentage of the total temperature difference between inside and outside air. This percentage may also be expressed as a ratio of heat resistances. In the case of a single glass window in which we omit the heat resistance of the glass as suggested by the author this ratio in terms of heat resistance would be $1/f_1 : 1/U$ which is the same as the ratio of the temperatures as suggested by the

author. Since the value of U depends upon the outside wind velocity which in practice varies through a wide range, the ratio R will also have a wide range of values depending upon the particular wind conditions. As two or more thicknesses of glass are used, with air spaces between, the effect of outside wind velocity on the overall coefficient becomes of less importance and the variation in the values of R would likewise be reduced for different wind velocities.

The author raises the question as to the accuracy of surface coefficients given in THE GUIDE. It should be pointed out that surface coefficients vary somewhat for normal building materials. The values of 1.65 for f_i and 6.00 for f_o have been selected as average values which may be applied for those building materials which do not possess high reflective coefficients such as bright metallic foils, etc. Surface coefficients were determined for several different materials in a cooperative research[1] between the Society and the University of Minnesota. In this work the coefficients for glass in still air at 40 F mean temperature and at 20 F mean temperature with a 15 mph wind velocity were found to be approximately 1.5 and 5.1 respectively. These values may be taken as the inside and outside surface coefficients for glass with a 15 mph wind velocity. If these coefficients are used and the glass is assumed to be ⅛ in. thick with a conductivity of 5.0, the overall coefficient U becomes 1.125, which is practically the same as the value of 1.13 given in THE GUIDE for single glass. Using these values the ratio R for a 15 mph outside wind becomes 0.75 and for still air it becomes 0.49.

R. E. HATTIS: I should like to ask if the windows were protected. Were they weather-stripped? I notice that double glazing is considered in connection with steel sash. My experience has been that steel sash condenses moisture even ahead of the single glass. What have your experiments indicated?

MEMBER: I wonder whether the authors have taken into account the question of glass sizes and the effect of heat transfer through those glasses. The size of the glass in question seems to have a great bearing upon the total heat transfer and temperature of the glass.

MR. RANDALL: I had hoped that I would have the opportunity of checking or discussing our conclusions with reference to the surface coefficients with Professor Rowley and those who are particularly interested in that item. I merely make the point that from those surface coefficients now in THE GUIDE, a value of R is obtained which is different from what we have obtained experimentally. Values of surface coefficients and U could be determined based on these revised values of R if discrepancies of the magnitude indicated in this paper would make consideration of revisions worthwhile. From the values we obtained there is simply evidence that would indicate a discrepancy.

With reference to the question of storm windows, the windows were loosely fitted, both single and double except when we used the effect of the wind, at which time all forms of windows had the cracks blocked.

This paper refers particularly to glass temperatures, which was the factor primarily under consideration in this investigation. The value of U was more or less a side issue to the proposition that we were really after, which was the condition under which deposition of moisture occurred.

So far as double glazing of steel sash is concerned, the deposition of moisture of steel windows, single glazed, occurs after the deposition of moisture on the glass, but not much later, and deposition of moisture on the double glazed steel window would certainly occur on the steel ahead of the double glass.

So far as glass sizes are concerned, the specimens which we used were steel windows with small panes, typical of the residential type of window. The wood win-

[1] Surface Conductances as Affected by Air Velocity, Temperature and Character of Surface, by F. B. Rowley, A. B. Algren and J. L. Blackshaw. A. S. H. V. E. TRANSACTIONS, Vol. 36, 1930.

dows had larger glass sizes. It is quite possible there are some discrepancies between small and large glass sizes. Inside surface temperature, however, did not vary with different glass sizes, other conditions being equal.

By reference to Table 1, it will be noted that the readings varied for various locations on the windows. In these investigations we were primarily concerned with an approximate value and did not take into consideration the slight plus or minus variations.

In so far as double glazing being a factor in the determination of the ratio, there was not much difference between double glazing and double windows. It varies in slight degree, as the charts show. Practically speaking, however, double glazing has some objections. First of all, some difficulty is encountered in selecting the same size of flat glass, and in addition, obtaining thin glass that has clear vision. The problem of sealing the glass is also an important factor, all of which presents many difficulties that make the product thus far rather expensive. Generally double windows can often be supplied cheaper than an arrangement of double glass.

The breathing of the inside humid air into the space between the double glass cannot seem to be entirely eliminated. In other words, the difference of the expansion of the outside light and the expansion of the inside light, one against the warm and one against the cold air, is sufficiently different to break the sealing bond and permit breathing. Then if the breathed in humid air cools enough to drop below the dew point, moisture results between the lights and the water cannot be easily removed.

There is one form of double glazing available which allows for breathing in of air from the outside through some sort of a pad of felt which restricts the entrance of outside dust. Whether this design is feasible to use, compared with a form of double window, has not been commercially demonstrated.

PROFESSOR ROWLEY: I should like to ask Mr. Randall whether a record of the wind velocities over the surface of the glass was observed during the tests. I do not note such a record in the paper. Wind velocities will vary the surface coefficients from 1.5 for still air up to 9.5 for 35 mph wind and would cause a variation in the ratio R of from 0.84 to 0.49. It is important that the wind velocities be carefully controlled and measured when determining any coefficient of this nature.

MR. RANDALL: We recognize in the paper that possibly the wind velocity was not 15 mph where it states, "Possibly the wind effect upon the outside glass surface in these tests was less than that previously determined as produced by a 15 mph wind, which accounts for the smaller value of R."

This statement was made in order to leave the question entirely open for discussion. In these investigations it was attempted to obtain the effect of a 15 mph wind, but this is a difficult matter to be sure of in a cold box, and it is a thing that we are of the opinion most investigators find difficult to correctly evaluate.

No. 1034

STUDY OF SUMMER COOLING IN THE RESEARCH RESIDENCE USING WATER FROM THE CITY WATER MAINS

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PREVIOUS investigations in summer cooling in the Research Residence during the summers of 1932,[1] 1933,[2] and 1934[3] made use of either ice as the medium for cooling water circulated through coils placed in the forced-air heating system or a two-ton mechanical refrigerating unit used in connection with evaporator coils placed in the forced-air heating system. These investigations were confined largely to studies of certain factors affecting the cooling load, and to studies of the effectiveness of different methods of circulating air from the outdoors at night, both as a supplement to artificial cooling during the day and as a means for eliminating the necessity for artificial cooling during the day. The investigation for the summer of 1935 was undertaken to determine to what extent water from the city water mains, available at a temperature of from 58 to 60 F, could be used to produce satisfactory cooling and dehumidification in the Research Residence when supplemented by the circulation of outdoor air through the second story at night and when approximately one air change per hour of outdoor air was used for the purpose of ventilation during the day.

Description of the Research Residence and Cooling Equipment

The Research Residence, shown in Fig. 1, together with the forced-air heat-

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[1] A. S. H. V. E. Research Paper, Study of Summer Cooling in the Research Residence at the University of Illinois, by A. P. Kratz and S. Konzo, A. S. H. V. E. Transactions, Vol. 39, 1933, p. 95.

[2] A. S. H. V. E. Research Paper, Study of Summer Cooling in the Research Residence for the Summer of 1933, by A. P. Kratz and S. Konzo, A. S. H. V. E. Transactions, Vol. 40, 1934, p. 167.

[3] A. S. H. V. E. Research Paper, Study of Summer Cooling in the Research Residence for the Summer of 1934, by A. P. Kratz, S. Konzo, M. K. Fahnestock and E. L. Broderick, A. S. H. V. E. Transactions, Vol. 41, 1935.

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ing system has been described in a previous paper.[4] For the purpose of this investigation the Residence was equipped with awnings at all east, south, and west windows, and the sun-room was isolated from the rest of the house by closing the doors leading into the dining room. The entire third story was regarded as an attic, and during the daytime was isolated from the rest of the house by closing the door at the head of the stairs. The attic windows, however, were opened to provide ventilation in the attic both day and night. With the exception of the space above the northwest bedroom and the small spaces adjacent to the dormer windows, the third story had hardwood floors laid on pine sub-flooring. The small spaces adjacent to the dormer windows had no floors. In the space above the northwest bedroom 1 in. of insulating blanket

FIG. 1. VIEW OF RESEARCH RESIDENCE IN URBANA, ILL.

was nailed to the upper edges of the floor joists. Hence, approximately all second floor ceilings were at least equivalent to lath and plaster with flooring above it. Practically no cooking was done in the kitchen, but the heat transmitted through the glass doors from the sunroom, which was not ventilated by opening the windows, compensated for this to a certain extent. Unless otherwise specified, the state of the Residence was the same for the work done during the four summers.

The arrangement of the forced-air duct system and fan is shown in Fig. 2. For the purpose of this investigation, all return ducts with the exception of the central one containing the cooling coil were blocked. The delivery ducts to the sun-room and third story were blocked, as indicated, and the dampers in the ducts to the first and second stories were adjusted to maintain the proper balance between the amounts of cooling required on these two stories.

The arrangement of the cooling plant is shown in Fig. 3. The cooling coil

[4] Loc Cit. See Note 1.

consisted of 8 rows of finned tubes and one row of tubes without fins. Each row consisted of 12 tubes connected into headers or return bends at the ends of the tubes. These headers were connected in such a way that the water flowed across the duct through one row of twelve tubes and returned through the next row, thus alternating through each succeeding row of tubes. The water connections were made so that the flow of the water was counter to the flow of the air, thus providing that the coldest air came in contact with the tubes containing the coldest water. The coil was arranged so that the air flowed horizontally. The water from the city service main passed through a calibrated water meter before entering the coil. As shown in Fig. 3 the tubes were grouped in sections of one, two, or three rows, and valves were provided

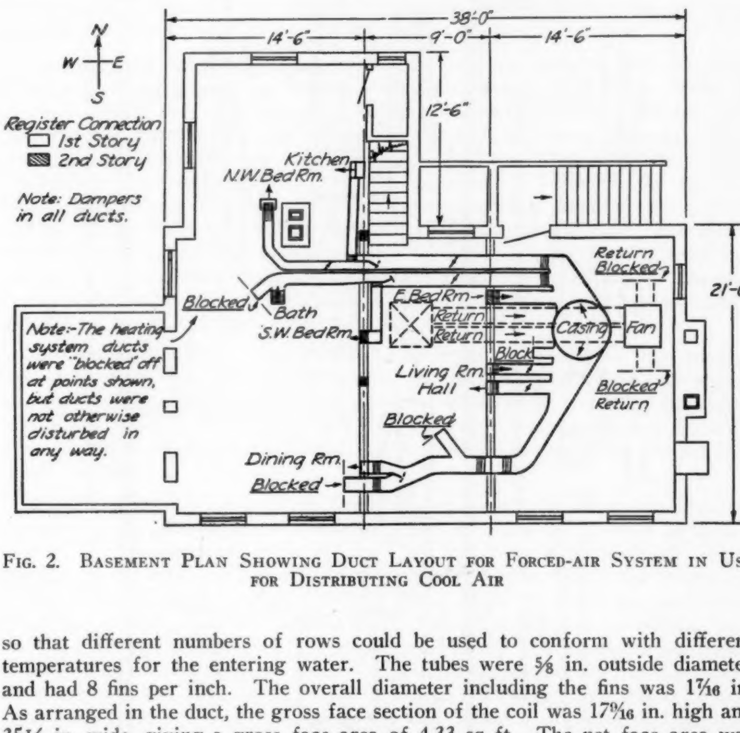

FIG. 2. BASEMENT PLAN SHOWING DUCT LAYOUT FOR FORCED-AIR SYSTEM IN USE FOR DISTRIBUTING COOL AIR

so that different numbers of rows could be used to conform with different temperatures for the entering water. The tubes were $\frac{5}{8}$ in. outside diameter and had 8 fins per inch. The overall diameter including the fins was $1\frac{7}{16}$ in. As arranged in the duct, the gross face section of the coil was $17\frac{9}{16}$ in. high and $35\frac{1}{2}$ in. wide, giving a gross face area of 4.33 sq ft. The net face area was 3.56 sq ft. The net free area was 1.76 sq ft. The total area of cooling surface, including the row without fins was 429 sq ft, and based on a volume of air circulated of 1277 cfm the gross face velocity was 295, the net face velocity was 359 and the velocity through the net free area was 726 fpm.

The selection of the cooling coil was based on the following considerations. Previous tests with a mechanical refrigerating unit in the Research Residence

(Table 1, 1934) [5] had indicated an actual maximum cooling load of approximately 30,000 Btu per hour, of which 22,000 was sensible heat and 8000 was latent heat load. Since, in the case of cooling with water, no additional load

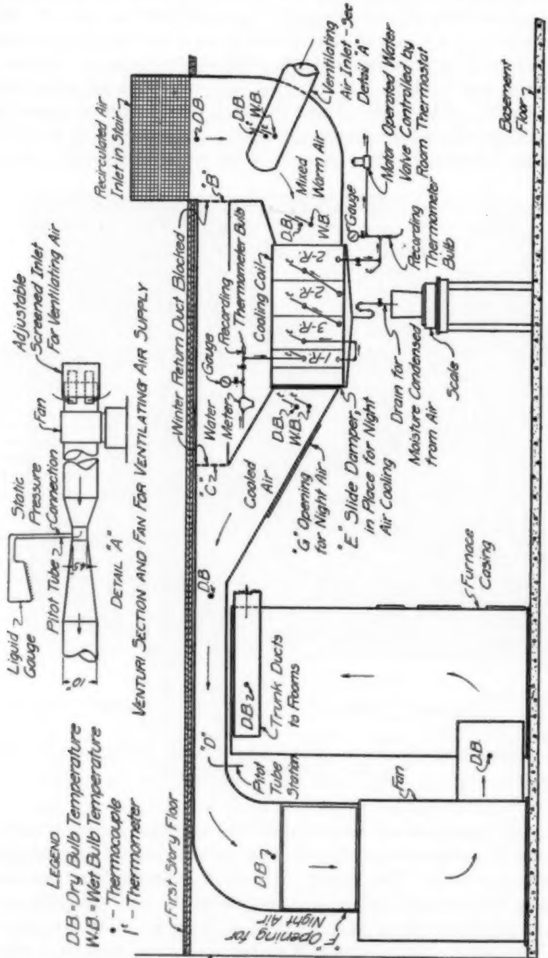

Fig. 3. Diagram of Cooling Plant with Water Cooling Coil

corresponding to the heat loss from the condensing unit and motor would be imposed, it was considered that these figures could be reduced approximately 10 per cent, and a total cooling load of 27,000 Btu per hour consisting of 20,000 Btu sensible heat and 7000 Btu latent heat was assumed as a basis for

[5] Loc. Cit. See Note 3.

the selection of the water coils. It was further considered desirable to operate the house under the same conditions as before with a room temperature of 80 F dry-bulb and with the air leaving the registers at approximately 70 F. Since the temperature rise in the ducts between the coil and the registers had been found to be approximately 4 F, the air was required to leave the coil at a temperature of 66 F. The admission of one air change per hour from outdoors to serve as ventilating air raised the temperature of the air entering the coils to approximately 81 F. The probable temperature drop through the coil would therefore be 15 F. From this temperature drop and the 20,000 Btu per hour, the amount of air to be circulated was calculated as 5560 lb per hour or approximately 1300 cfm. The temperature of the available service water was approximately 60 F, and a rise of 10 F in the water passing through the coil was assumed. From this rise and the total cooling load of 27,000 Btu per hour, the probable water requirement was determined as 324 gal per hour.

From the manufacturer's catalog a stock coil unit was selected such that with 1300 cu ft of air flowing per minute a net face velocity of 365 fpm would be obtained. With this unit the water velocity was estimated as 0.556 fps, and from the temperatures of the entering and leaving air and water, the logarithmic mean temperature difference was calculated as 8.26 F. Following the manufacturer's recommendation, the heat transmission was based on the sensible heat alone. The overall heat transmission coefficient of 5.60 Btu per square foot per degree mean temperature difference per hour, corresponding to a water velocity of 0.556 fps and an air velocity of 365 fpm was selected from the manufacturer's published data. The sensible cooling load of 20,000 Btu per hour required 433 sq ft of coil surface and this condition was approximately satisfied by 8 rows of tubes in the type of coil used.

The operation of the cooling plant was controlled by means of a thermostat located at the 60-in. level in the hall on the second story. This thermostat was used in connection with a motor-operated, water-valve placed on the outlet side of the cooling coil to start and stop the flow of water through the coil. The valve was either open or closed and moved slowly enough to prevent any water hammer.

In order to provide for both heating in the winter and cooling in the summer, the cooling coil was installed in a by-pass in the central return duct as shown in Fig. 3. For the summer work, the duct was blocked with tightly fitting dampers at *B* and *C*, and all of the air delivered by the fan in the forced-air heating system passed through the cooling coil when the air in the house was being recirculated. For the purpose of providing outdoor air for cooling during the night, a slide damper, *E*, was placed in the by-pass on the down-stream side of the coil, and a door, *G*, was placed in the recirculating duct on the down-stream side of the slide damper. For a few of the earlier tests this door was located as shown at *F* in Fig. 3, but its use was discontinued owing to difficulties in measuring the air delivery from the fan. When outdoor air was required, the basement door and the door, *G*, in the recirculating duct were opened, and the slide damper, *E*, was closed. The fan in the forced-air system was driven by a V-belt from a ¾ hp motor and delivered approximately 1300 cu ft of air per minute when recirculating the air in the house and 2300 cfm when using outdoor air at night. Outdoor air, for the purpose of ventilation during the day, was provided by means of the duct shown as Detail *A* in Fig. 3. This duct contained a venturi section for the measurement of the volume of air delivered,

and it was necessary to use a small auxiliary fan in order to deliver the equivalent of approximately one air change per hour. Wet- and dry-bulb temperatures of the air entering and leaving the cooling coil were measured by means of thermometers. Two pairs of these thermometers were placed on each side of the coil, a pair near each side of the duct. The bulbs were located about 5 in. from the sides of the duct, and at about the middle of the height of the cross-section. As a check on these thermometers, two hair hygrometers were also located in the duct in cross-sections corresponding to the locations of the thermometers. The temperature of the water at inlet, at outlet and at intermediate points was measured by means of thermometers as shown in Fig. 3. The locations of thermometers and thermocouples for measuring the temperature of air at other points and the location of Pitot tubes for measuring the air quantities are also shown in this figure.

METHOD OF CONDUCTING TESTS

During the summer continuous records were made, by means of temperature recorders, of the following air temperatures: outdoor, living room, dining room, kitchen, first, second and third story halls, east bedroom, southwest bedroom, and northwest bedroom. Continuous records were also made of the temperatures of the air entering and leaving the cooling coil, the water entering and leaving the cooling coil, the outdoor air taken in for ventilation and the wet- and dry-bulb temperature of the outdoor air on the north side of the house. Other incidental air temperatures were read at regular intervals. Relative humidities both indoors and outdoors were observed by means of an aspirating psychrometer. The outdoor wet- and dry-bulb readings on this psychrometer served as a basis for correcting all outdoor wet- and dry-bulb temperatures read from the recorder charts.

During the periods of operation, observations were made of the weight of water circulated through the cooling coil, the temperature of the water entering and leaving the coil, the weight of water condensed from the air and the electrical input to the fan motor. The weight of water circulated was obtained by means of a calibrated water meter and the air quantities were obtained from traverses made with Pitot tubes at section D in the central return duct and at the venturi section in the ventilating air duct as shown in Fig. 3.

During all of the tests the windows on the first story remained closed. The windows in the attic, with the exception of one opposite the door at the top of the stairs, remained open. For the purpose of cooling with outdoor air at night, 11 windows on the second story were opened by raising the lower sash to the full extent. Two windows which were opposite registers and one window at the second floor stair landing remained closed. For most of the tests the Residence was operated on the schedule as given:

The second story windows and the attic and basement doors were closed at 7 a.m. and the dampers were changed so that the fan, which had been delivering outdoor air through the system, started delivering recirculated air and the outdoor air admitted for ventilation. The former was equivalent to 4.4 recirculations of the air in the house, and the latter was equivalent to one air change per hour, making a total of 5.4 air changes per hour, delivered by the fan. The fan was allowed to run continuously during the day. When the temperature of the indoor air on the second story rose to 81 F the motor driven valve

actuated by the thermostat admitted water to the cooling coil. The cooling plant was allowed to operate with thermostatic *on* and *off* control maintaining 81 F on the second story until the effective temperature outdoors became equal to the effective temperature on the second story indoors. The water valve was then closed; the second story windows and attic door were opened; and the dampers were set and the basement door opened, so that the fan delivered outdoor air through the duct system, the fan continuing to run until 7 a.m. The fan delivery was 2300 cfm or 9.74 air changes per hour.

TABLE 1. TYPICAL OPERATING DATA AND RESULTS OBTAINED BY COOLING WITH WATER, AT 2:00 P.M. ON AUGUST 2, 1935[a]

1. Outdoor air	D.B.[b] 95.0 F, W.B. 78.6 F, D.P. 72.5 F
Humidity	R.H. 48.5%, S.H. 120.8 grains per pound dry air
2. Indoor air, aver. breathing level temperature	1st story 79.1 F, 2nd story 80.5 F
3. Indoor air, house aver. at breathing level	D.B. 79.8 F, W.B. 70.9 F, D.P. 67.2 F
Humidity	R.H. 65%, S.H. 99.8 grains per pound dry air
4. Indoor air, entering return grille	D.B. 77.7 F, W.B. 69.8 F, D.P. 66.2 F
Humidity	R.H. 67.5%, S.H. 96.8 grains per pound dry air
5. Ventilating air	D.B. 91.6 F, W.B. 76.2 F, D.P. 70.1 F
Humidity	R.H. 49.5%, S.H. 111.0 grains per pound dry air
6. Mixed air entering cooling coil	D.B. 80.3 F, W.B. 70.7 F, D.P. 66.5 F
Humidity	R.H. 62.8%, S.H. 97.5 grains per pound dry air
7. Mixed air leaving cooling coil	D.B. 65.8 F, W.B. 64.7 F, D.P. 64.1 F
Humidity	R.H. 94.0%, S.H. 89.7 grains per pound dry air
8. Air temperature drop through cooling coil, F	14.5
9. Temperature of cooled air leaving registers, 1st and 2nd story aver., F	70.2
10. Air temperature rise in ducts and casing, F	4.4
11. Basement air temperature at breathing level, F	75.2
12. Quantity of air circulated through wet coils	1,277 cfm or 5,547 lb of dry air per hour
Density of air, pounds per cubic foot	0.0724
13. No. of house air recirculations per hour	5.4
14. Ventilating air	238 cfm or 1005 lb of dry air per hour
Density of air, pounds per cubic foot	0.0704
15. Cooling coil	gross face area 4.33 sq ft; net free area 1.76 sq ft
Surface of cooling coil, square feet	429
Air velocity, feet per minute	gross face 295, net face 359, free area 726
16. Moisture condensed from air, pounds per hour	5.83
17. Heat given up by the air; total	25,440 Btu per hour
Heat due to moisture in air	6,120 Btu per hour; 24.0% of total heat absorbed
Sensible heat	19,320 Btu per hour; 76.0% of total heat absorbed
18. Water temperature through cooling coil, F	Inlet 58.3, Outlet 66.2, Rise 7.9
19. Rise in water temperature through coil, F	One row 0.8, Four rows 4.2, Six rows 5.5, Eight rows 7.9
20. Water pressure at coil, pounds per square inch	Inlet 20.5, Outlet 20.0
21. Quantity of cooling water per hour	Pounds 3000.6, Gallons 360
22. Heat absorbed by water passing through cooling coil, Btu per hour	23,700
23. Recirculating fan data	Speed 477 rpm
Motor	Size ¼ hp, Measured power rate 0.44 kw
24. Static pressure loss through coils	Wet 0.28 in. water, Dry 0.23 in. water
25. Total resistance of system	0.51 in. water

[a] Test number 13, series 1–35.
[b] Abbreviations used: D.B. = Dry-Bulb temperature; W.B. = Wet-Bulb temperature; D.P. = Dew Point temperature; R.H. = Relative Humidity; S.H. = Specific Humidity.

During the latter part of the season this schedule was modified by maintaining an effective temperature of approximately 74 F instead of a dry-bulb temperature of 81 F. This was accomplished by controlling the plant manually and starting the flow of water through the coil whenever the relative humidity increased sufficiently so that with the prevailing dry-bulb temperature the effective temperature rose above 74 F.

RESULTS OF TESTS

General Conditions

The operating characteristics of the cooling plant and a comparison of the actual and calculated cooling loads for the house can best be illustrated by the

TABLE 2. COMPLETE DATA AND RESULTS FOR TESTS

Date 1935	Test No.	Series No.	Weather Data: Degree-Hours Above 85 Deg	Weather Data: Degree-Hours Above 90 Deg	Weather Data: Sun Hour: Min.	Outdoor Air: Max. Temp. During Test F	Outdoor Air: Min. Temp. Previous to Test F	Outdoor Air: Average During Period Requiring Cooling with Water Temp. F	Outdoor Air: Average During Period Requiring Cooling with Water Hum. %	Indoor Air Conditions: At End of Night Cooling Temp. F	Indoor Air Conditions: Average During Period Requiring Cooling with Water Temp. F	Indoor Air Conditions: Average During Period Requiring Cooling with Water Hum. %	Temperature Difference: Outdoor Max. to Average Indoor F	Temperature Difference: Average Outdoor to Average Indoor F	Operation of the Fan: Night Air Cooling Previous to Test Duration Hours	Operation of the Fan: Night Air Cooling Previous to Test Power Kwhr	Operation of the Fan: Recirculation Previous to Test Duration Hours	Operation of the Fan: Recirculation Previous to Test Power Kwhr	Operation of the Fan: Recirculation During Test Period Duration Hours	Operation of the Fan: Recirculation During Test Period Power Kwhr
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21
July 18	2	I–35	65.20	20.56	14:45	94	67	87.8	55.6	73.0	79.3	62.7	14.7	8.5	11.50	6.3	9.05	3.3	4.78	2.0
19	3	I–35	32.00	4.64	8:20	92	70	89.4	49.3	76.0	79.8	64.5	12.2	9.6	10.17	5.4	7.45	2.8	2.55	1.1
21	4	I–35	7.84	0	11:06	87	69	82.8	76.7	73.7	78.9	70.6	8.1	3.9	9.33	5.1	7.68	3.8	4.65	1.7
22	5	I–35	2.56	0	10:10	86	73	83.9	71.5	76.1	79.4	70.2	6.6	4.5	9.75	5.2	7.63	2.9	5.28	2.0
25	6	I–35	19.36	0	12:54	89	75	87.7	64.7	77.0	79.7	67.4	9.3	8.0	11.92	6.0	6.62	2.6	2.88	1.3
26	7	I–35	11.36	0	14:32	89	70	85.5	66.0	75.7	79.8	68.0	9.2	5.7	14.50	8.7	8.20	3.5	4.80	2.1
27	8	I–35	15.92	0	12:49	88	69	83.6	71.2	75.4	78.8	68.9	9.2	4.8	11.00	6.6	6.58	2.8	7.92	3.6
28	9	I–35	36.40	3.92	11:45	92	74	86.9	80.9	76.2	79.3	70.1	12.7	7.6	9.50	5.5	6.93	3.2	8.57	3.8
30	10	I–35	60.64	16.40	14:24	94	65	88.0	68.7	73.2	79.9	67.3	14.1	8.1	10.50	6.2	8.68	3.8	6.32	2.8
31	11	I–35	71.20	22.96	12:05	95	76	88.9	68.1	79.7	79.5	66.4	15.5	9.4	9.00	5.3	2.08	0.8	14.42	6.6
Aug. 1	12	I–35	80.96	29.12	11:20	96	74	89.0	64.0	78.3	79.7	68.2	16.3	9.3	7.50	4.5	2.85	1.3	14.00	6.4
2	13	I–35	84.32	31.92	13:40	96	76	89.8	61.7	78.8	79.7	68.1	16.3	10.1	7.15	4.1	1.85	0.9	14.50	6.8
3	14	I–35	8.40	0	12:46	88	74	84.4	71.8	77.5	79.7	77.2	8.3	4.7	7.65	4.4	4.72	2.2	8.70	4.0
4	15	I–35	34.88	1.12	14:14	91	69	86.5	58.6	77.5	79.8	63.2	11.2	6.7	10.50	6.3	8.25	3.7	4.25	2.0
5	16	I–35	26.24	2.32	9:13	91	70	76.5	87.8	75.2	80.2	74.0	10.8	—3.7	11.50	6.7	6.73	3.0	3.77	1.8
6	17	I–35	65.52	19.12	14:09	94	72	87.2	71.3	75.9	79.2	69.3	14.8	8.0	13.50	8.4	7.12	3.3	9.38	4.4
7	18	I–35	1.92	0	13:21	86	76	84.6	71.6	77.9	80.0	75.1	6.0	4.6	7.50	4.7	3.65	1.8	7.85	3.5
9	19	I–35	18.00	0	12:46	88	68	81.6	79.1	73.2	80.6	71.5	7.4	1.0	12.00	7.5	11.82	5.5	2.00	1.0
10	20	I–35	14.96	0	11:05	89	70	85.7	65.1	76.2	80.1	71.2	8.9	5.6	10.18	6.2	8.50	4.1	4.25	1.9
11	21	I–35	46.88	5.44	13:59	92	70	87.4	66.6	75.5	79.8	67.1	12.2	7.6	11.25	7.0	6.90	3.2	7.60	3.6
12	22	I–35	35.04	4.40	14:26	92	72	87.2	58.4	76.4	80.3	69.0	11.7	6.9	9.50	6.0	4.78	2.1	6.72	3.1
15	23	II–35	0	0	13:30	82	68	80.6	75.5	72.7	76.1	73.3	5.9	4.5	13.50	7.6	8.40	3.9	3.10	1.3
16	24	II–35	3.20	0	13:27	87	68	83.2	70.5	73.2	76.3	81.0	10.7	6.9	12.50	6.3	3.43	1.6	8.57	3.6
18	25	II–35	2.48	0	13:04	86	68	84.1	72.6	73.9	76.5	78.7	9.5	7.6	12.25	6.0	4.58	2.0	6.17	2.7
19	26	II–35	2.80	0	8:30	86	70	84.2	72.0	74.3	76.4	83.5	9.6	7.8	13.00	6.5	3.73	1.7	8.27	3.5
20	27	II–35	1.12	0	10:56	86	70	83.3	76.8	74.5	76.7	77.6	9.3	6.6	12.00	5.9	6.37	2.8	8.10	3.5

Note: Test No. 1 was omitted on account of incomplete data.

TABLE 2. (*Continued*)

Date 1935	Test No.	Series No.	Overall Test Period: Start of Cooling with Water to Start of Night Air Cooling			Water Circulation Through Cooling Coil				Total Moisture Condensed Lb	Heat Absorbed by Cooling Coil During Overall Test Period, Btu		
			Start	End	Duration Hours	Time in Hours	Quantity in Gallons	Temp. at Inlet F	Temp. Rise Through Coil F		Load Due to Moisture	Sensible	Total
1	2	3	22	23	24	25	26	27	28	29	30	31	32
July													
18	2	I–35	4:04P	8:50P	4.78	4.77	1784	57.7	6.7	15.28	16,050	83,613	99,663
19	3	I–35	2:27P	5:00P	2.55	2.55	860	57.7	7.6	7.88	8,280	46,212	54,492
21	4	I–35	4:41P	9:20P	4.65	3.15	1173	58.3	8.4	22.30	23,420	58,732	82,152
22	5	I–35	2:43P	8:00P	5.28	4.03	1505	58.1	8.1	30.32	31,840	69,775	101,615
25	6	I–35	1:37P	4:30P	2.88	2.88	1048	58.2	8.3	19.47	20,450	52,092	72,542
26	7	I–35	3:12P	8:00P	4.80	4.10	1476	58.3	8.2	21.92	23,030	77,830	100,860
27	8	I–35	1:35P	9:30P	7.92	6.32	2253	58.1	7.8	32.62	34,270	112,214	146,484
28	9	I–35	1:56P	10:30P	8.57	6.53	2432	58.3	8.2	47.06	49,420	116,794	166,214
30	10	I–35	3:41P	10:00P	6.32	4.07	1481	58.2	8.9	23.67	24,860	84,966	109,826
31	11	I–35	9:05A	11:30P	14.42	12.10	4266	57.5	8.9	93.14	97,850	218,634	316,484
Aug.													
1	12	I–35	9:51A	11:51P	14.00	11.33	3997	58.2	8.2	66.06	69,350	203,874	273,224
2	13	I–35	8:51A	11:21P	14.50	11.93	4350	58.3	7.9	60.37	63,400	223,054	286,454
3	14	I–35	11:48A	8:30P	8.70	3.48	1276	58.3	9.1	10.45	10,970	85,854	96,824
4	15	I–35	3:15P	7:30P	4.25	3.20	1200	58.3	7.4	4.25	4,460	69,540	74,000
5	16	I–35	1:44P	5:30P	3.77	0.35	133	58.2	9.9	a	....	....	10,949
6	17	I–35	2:07P	11:30P	9.38	7.25	2711	58.5	8.3	50.78	53,350	134,230	187,580
7	18	I–35	10:39A	6:30P	7.85	1.80	704	58.8	8.9	a	....	....	52,243
9	19	I–35	6:49P	8:49P	2.00	0.38	147	58.6	9.1	a	....	....	11,160
10	20	I–35	3:30P	7:45P	4.25	1.82	667	58.5	8.6	a	....	....	47,816
11	21	I–35	1:54P	9:30P	7.60	3.92	1468	58.5	8.1	12.60	13,240	85,904	99,144
12	22	I–35	11:47A	6:30P	6.72	2.07	766	58.3	9.1	3.35	3,520	54,584	58,104
15	23	II–35	3:24P	6:30P	3.10	1.72	639	58.6	8.0	4.00	4,200	38,440	42,640
16	24	II–35	10:26A	7:00P	8.57	3.62	1390	58.3	8.0	11.45	12,030	80,690	92,720
18	25	II–35	11:50A	6:00P	6.17	3.20	1182	58.4	8.2	16.60	17,430	63,340	80,770
19	26	II–35	10:44A	7:00P	8.27	4.45	1651	58.3	7.9	26.23	27,570	81,134	108,704
20	27	II–35	1:22P	9:28P	8.10	5.22	1718	58.1	8.1	31.95	33,550	82,442	115,992

Test too short to permit accurate determination of weight of moisture condensed.

results obtained on a typical day. For this purpose a test made on August 2, 1935, was selected and the results are shown in Table 1 and Fig. 4. General results from all tests are given in Table 2.

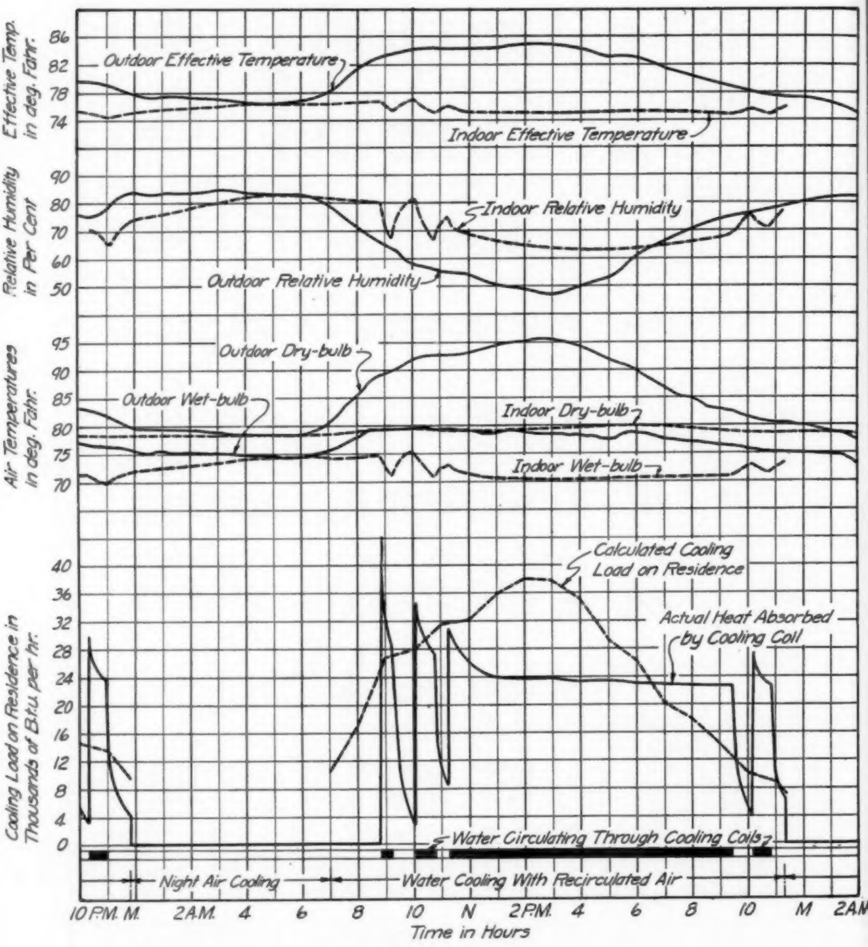

FIG. 4. ACTUAL AND CALCULATED COOLING LOAD ON RESIDENCE AND AIR TEMPERATURES, TEST NO. 13, SERIES 1-35, AUGUST 2, 1935; WITH NIGHT AIR COOLING

On the typical day selected, the outdoor temperature was 95.0 F at 2 p.m. and reached a maximum of 96.0 F at 2:30 p.m. At this time the house was being operated with a constant dry-bulb temperature of approximately 80 F,

and the windows on the second story had been opened and outdoor air circulated on the previous night. The air conditions at different locations are given in the first eleven items of Table 1. The dry-bulb temperature on the second story was only 0.6 F higher than that on the first, thus indicating a satisfactory balance of the cooling on the two stories. The temperature of the cooled air leaving the registers was approximately 70 F, and no difficulty was experienced from drafts in the rooms. The rise in temperature of the air passing through the furnace casing and ducts was 4.4 F. The quantity of air circulated was 1347 cfm when the coils were dry. Increased frictional resistance when condensation was present on the coil surfaces, however, decreased this amount to 1277 cfm. Since part of the coil surfaces were always wet during actual operating periods, all calculations were based on 1277 cfm for the volume of air circulated at the density of 0.0724 lb per cu ft. With this amount of air, which represented 5.4 recirculations per hour, the velocity through the net face area was 359 fpm and the temperature drop through the coil was 14.5 F. This corresponded closely with the 365 fpm and the 15.0 F drop used for the selection of the coils. The free area velocity of 726 fpm did not prove to be sufficient to carry any condensed moisture away from the surfaces of the cooling coil.

Cooling Load

Since the cooling coil was well lagged with corkboard to prevent heat gain, it was possible to obtain the actual cooling load either from the heat given up by the mixture of air and water vapor or from the heat absorbed by the water passing through the coil. Considerable difficulty, however, was experienced in obtaining a heat balance between these two quantities. The discrepancy was approximately 20 per cent. It was found, by making a complete traverse in the duct on both sides of the coil, that both the wet-bulb and dry-bulb temperatures varied from point to point in the cross-section of the duct. By comparing the results of these traverses with the averages of the temperatures obtained from the readings of the wet- and dry-bulb thermometers at the two reading stations on each of the upstream and downstream sides of the coil, it became evident that a correction of —0.1 F had to be added to the average dry-bulb reading on the upstream side and one of +1.0 F to that on the downstream side. Also a correction of +0.1 F had to be added to the average wet-bulb reading on the upstream side and one of +0.5 F to that on the downstream side. In addition to these corrections it was found necessary to make a correction of —0.4 F to account for radiation received by the wet-bulb thermometers on the upstream side. No radiation correction was necessary on the downstream side, since the air was very nearly saturated and the wet-bulb temperature was practically the same as the dry-bulb temperature of the air and surrounding surfaces. By using these corrected wet-bulb readings to determine the enthalpy of the moist air entering and leaving the coil it was possible to obtain a heat balance between the air and water within approximately 8 per cent. This method of calculation, however, does not show the distribution between sensible and latent heat loads. This distribution was obtained, as shown in Item 17, Table 1, by using the corrected values for the dry-bulb temperatures, and by multiplying the weighed amount of condensation from the coil by a constant representing the heat given up by the change in moisture content of the air per pound of water vapor condensed. This latter included the latent heat and superheat in the water vapor

condensed, and the change in the superheat of the water vapor remaining in the air after passing the cooling coil, and amounted to 1,050 Btu per pound of vapor condensed. The total of the heat given up by the air as computed by this method was within 6 per cent of the calculated heat absorbed by the water. For the purpose of analysis, and in the general results shown in Table 2, the cooling load calculated from the water circulated through the coil, as shown in Item 22, Table 1, was accepted as correct. The latent heat load was calculated from the weighed amount of moisture condensed from the air, and the sensible heat load was obtained by difference. On all tests having a duration exceeding 4 hours, the moisture load varied from 20 to 31 per cent of the total load.

It is not generally recognized that wet-bulb readings are subject to a radiation correction that can not be eliminated by any processes of shielding or increasing the velocity of the air passing over the instrument. This correction is a minimum in saturated air, where the wet-bulb depression is zero, and increases as the air becomes drier and the wet-bulb depression increases. Owing to this fact, and to point to point variations in both wet- and dry-bulb temperatures in the cross-section of a duct, it should be emphasized that heat load determinations based on the air side of a system alone should be accepted with extreme caution unless some independent check on the validity of the results is available.

OPERATING CHARACTERISTICS WITH CONSTANT DRY-BULB TEMPERATURE

The operating characteristics with a constant dry-bulb temperature in the rooms of approximately 80 F are shown in Fig. 4, for a day on which the outdoor temperature rose to a maximum of 96 F. During the night before, when the second story windows were opened and outdoor air was circulated, the indoor dry-bulb temperature dropped to 78 F and remained at this value until 7:00 a.m., when the windows were closed and the dampers were changed to recirculate the air in the house. The dry-bulb temperature rose to 79 F at 8:50 a.m. at which time the thermostat opened the water valve. The plant then operated intermittently until 11:20 a.m., and continuously from 11:20 a.m. to 9:30 p.m. maintaining the dry-bulb temperature between 79 and 80 F.

At 8:50 a.m. the wet-bulb temperature was 75 F, corresponding to a relative humidity of 80 per cent. When the plant started the relative humidity dropped to 67 per cent, but rose again to 81 per cent almost immediately after the water valve closed. This alternate decrease and increase of relative humidity in the house was characteristic of the intermittent operation of the plant. When the plant operated continuously, however, as from 11:20 a.m. to 9:30 p.m., the relative humidity stabilized at between 63 and 66 per cent, representing an effective temperature of approximately 75 F. During the intermittent periods the effective temperature was at times as high as 77 F. That is, on a warm day, when the plant operated continuously, conditions were just on the upper borderline for comfort, but during the periods of intermittent operation the effective temperature was distinctly too high for comfort. The significance of this type of operation on a milder day will be referred to later, but conditions shown in Fig. 4 indicated that for 58 F service water, with which it was not found possible to obtain a relative humidity lower than 62 per cent, the dry-bulb temperature should be maintained somewhat lower than 80 F.

The calculated cooling load was computed by the method outlined in a pre-

vious paper.[6] The calculated load was greater than the actual load during the period from 11:15 a.m. to 6:30 p.m., rising to a maximum value of 38,000 Btu per hour at 2:30 p.m. as compared with an actual load of 24,000 at the same time. After 6:30 p.m. the calculated load became less than the actual load. This tendency for the actual and calculated load curves to cross at approximately 6:30 p.m. was also noted on the previous work with ice cooling and with the mechanical refrigerating unit.

During the periods of intermittent operation the actual cooling load curve showed high peaks at the start of each operating period. These peaks correlated with periods of high indoor relative humidity, indicating that periods of high wet-bulb temperature for the air entering the coil were accompanied by

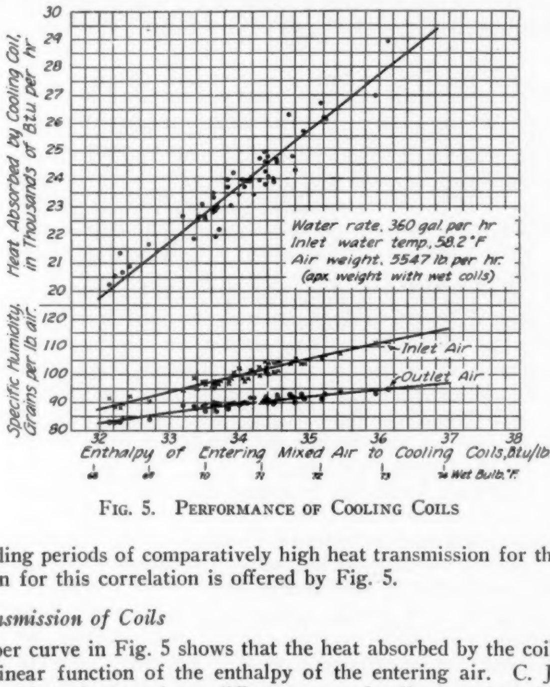

FIG. 5. PERFORMANCE OF COOLING COILS

corresponding periods of comparatively high heat transmission for the coil. An explanation for this correlation is offered by Fig. 5.

Heat Transmission of Coils

The upper curve in Fig. 5 shows that the heat absorbed by the coil was practically a linear function of the enthalpy of the entering air. C. J. Scanlan[7] has shown this to be true for a different type of coil when the velocity of the air and the mean temperature of the refrigerant in the coil were maintained constant. The deviations of the points from the mean curve shown in Fig. 5 can therefore be explained by variations in mean water temperature, and in the weights of air and water circulated under the conditions of the tests. The maximum deviation shown, however, was only 6 per cent.

During these tests a constant indoor dry-bulb temperature was maintained,

[6] Loc. Cit. See Note 3.
[7] Performance of Extended Cooling Surfaces, by C. J. Scanlan, *Refrigerating Engineering*, Vol. 27, No. 4, April, 1934, pp. 197-199.

resulting also in a practically constant dry-bulb temperature for the air entering the coil. Hence, periods of high relative humidity also represented periods of high wet-bulb temperature with corresponding high enthalpy for the air entering the coil; and since Fig. 5 shows that high heat transmission accompanied high enthalpy for the air entering the coil, it also serves to explain the fact that the peaks in the load curve in Fig. 4 occurred at times corresponding to high indoor relative humidity.

Although the same type of curve as that shown in Fig. 5 also applies to the evaporator coil used in connection with the mechanical refrigerating unit for the studies in the summer of 1934, this plant did not show the high peaks in the load curve which were characteristic of the water cooling plant. In the case of the evaporator coil the heat transmission at the time of starting was limited by the amount of refrigerant that could enter the coil. This refrigerant would

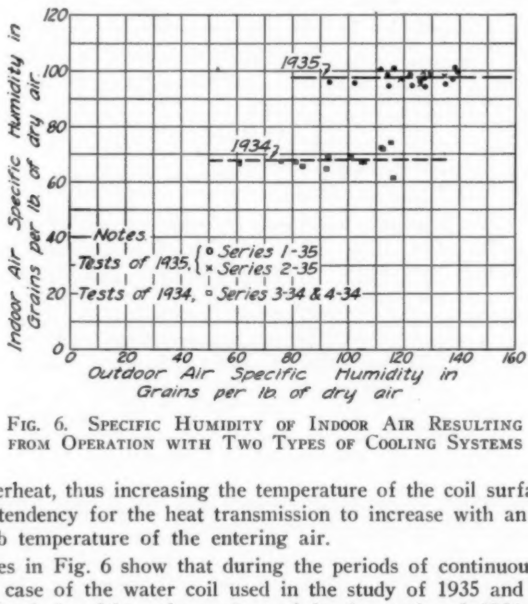

FIG. 6. SPECIFIC HUMIDITY OF INDOOR AIR RESULTING FROM OPERATION WITH TWO TYPES OF COOLING SYSTEMS

tend to superheat, thus increasing the temperature of the coil surface and offsetting the tendency for the heat transmission to increase with an increase in the wet-bulb temperature of the entering air.

The curves in Fig. 6 show that during the periods of continuous operation, both in the case of the water coil used in the study of 1935 and in the case of the mechanical refrigerating unit used in the study of 1934, the indoor specific humidity maintained was dependent only on the temperature of the medium used and the character of the cooling coil and was independent of the amount of water vapor in the outdoor air. Since in both cases a constant indoor dry-bulb temperature of 80 F was maintained, the specific humidity is also representative of the indoor relative humidity. That is, the water coil used in 1935 with a mean water temperature of approximately 62 F maintained a constant indoor relative humidity of 63 per cent during periods of continuous operation, while the evaporator coil used in 1934 with a mean temperature of approximately 45 F for the refrigerant maintained a constant relative humidity of 45 per cent.

During periods of operation the ventilating air from outdoors was mixed with the recirculated air from the house just before entering the cooling coil. The moisture content of the outdoor air therefore directly influenced that of the mixed air entering the coil. The lower curves in Fig. 5 indicate that the enthalpy of the entering air increased as the specific humidity increased, resulting in an increase in the heat transmission for the coil. At the same time the specific humidity of the air leaving the coil increased, but not to as great an extent as that of the inlet air. This indicates that a part, if not all, of the increase in heat transmission was utilized in increasing the amount of condensation as the specific humidity of the outdoor air, and hence that of the air entering the coil, was increased. Thus the increasing heat transmission of the coil tended to compensate for increases in the outdoor specific humidity, and, as a result, also tended to maintain a constant indoor relative humidity during periods of continuous operation, as shown in Fig. 6.

OPERATING CHARACTERISTICS WITH CONSTANT EFFECTIVE TEMPERATURE

Fig. 7 shows the performance of the Residence on two mild days, during which the operation of the plant was intermittent and no long periods of continuous running occurred. The left hand side shows the operation on a day on which the outdoor temperature rose to a maximum of 87 F and a constant indoor dry-bulb temperature of approximately 80 F was maintained. Owing to the short operating periods, the indoor relative humidity ranged from 68 to 80 per cent with corresponding indoor effective temperatures of from 76 to 77.5 F. At no time could the conditions be regarded as comfortable. A study was therefore made on a similar day on which the maximum outdoor temperature rose to 86 F and during which the plant was manually controlled to maintain an indoor effective temperature not to exceed 76 F. The results of this study are shown in the right hand side of Fig. 7. The second story windows were closed at 7:00 a.m. and at 10:00 a.m. the indoor dry-bulb temperature was 75 F and the relative humidity was 90 per cent, corresponding to an effective temperature of 74 F. The plant was then started and the relative humidity decreased to 74 per cent, giving an effective temperature of 72 F at 11:00 a.m., at which time the plant was stopped. Subsequent to this the plant was started manually whenever the effective temperature approached 74 F with the prevailing dry-bulb temperature and was stopped whenever the effective temperature dropped to 73 F. With this method of operation the indoor dry-bulb temperature gradually increased to 77 F and the relative humidity decreased to vary between 70 and 80 per cent during the on- and off-periods of operation. By this means comparatively comfortable conditions were maintained in the house during the entire day.

The results of these studies indicate that it would be desirable to develop a means of control operating directly on effective temperature. With 58 F service water available, and with a constant dry-bulb temperature of 80 F maintained, on warm days, when the maximum outdoor temperature rose above 90 F, and when the plant could operate continuously, a material improvement in atmospheric conditions was effected that would be acceptable to the majority of users, even though optimum conditions were not obtained. On mild days, however, this could not be accomplished with a constant dry-bulb temperature of 80 F, but by setting the thermostat to maintain approximately 76 F acceptable

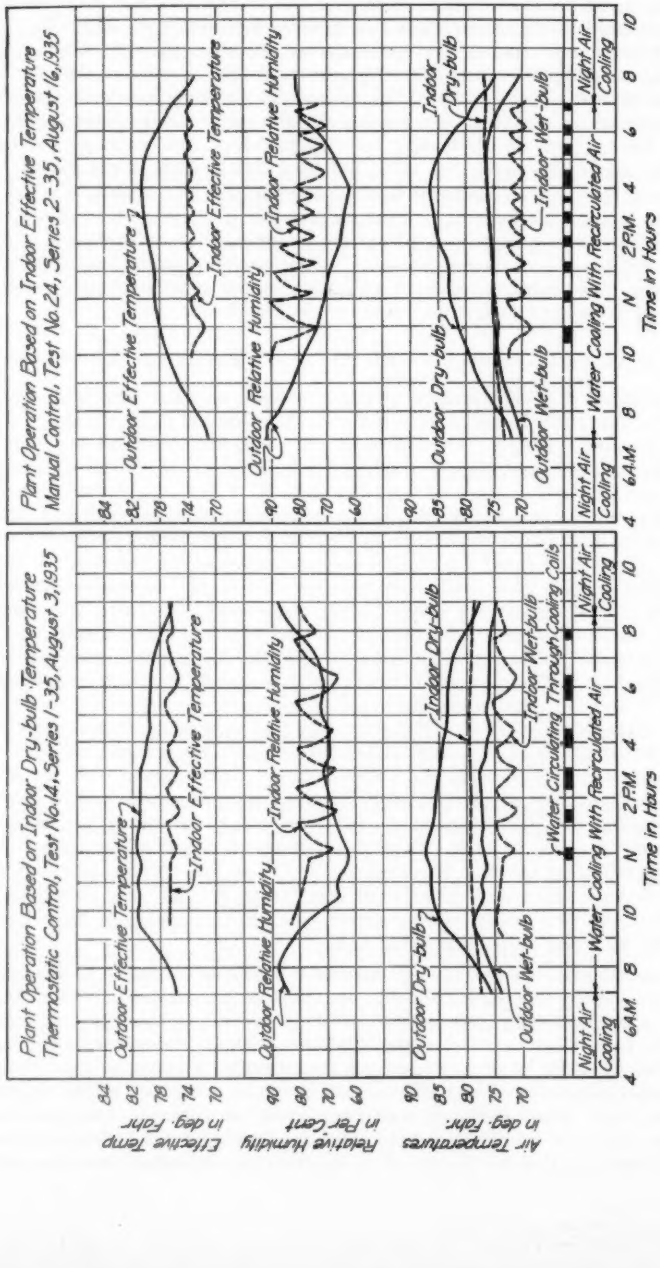

FIG. 7. INDOOR AND OUTDOOR TEMPERATURE CONDITIONS WITH COOLING PLANT OPERATION BASED ON INDOOR DRY-BULB TEMPERATURE AND ON INDOOR EFFECTIVE TEMPERATURE

conditions could be maintained. The results further indicated that 60 F is probably the upper practical limit for the temperature of the available cooling water. With higher initial water temperatures the amount of dehumidification or reduction in indoor relative humidity would be very small. Consequently, room temperatures considerably lower than 80 F would have to be maintained and the amount of coil surface required would be excessive.

Factors Affecting the Selection of Cooling Coils

In selecting the cooling coils required to absorb the maximum cooling load on the Research Residence, it was found that there was considerable latitude in the choice, provided no restrictions were placed on the amount of cooling water or the amount of coil surface used. The cooling load was estimated at 20,000 Btu per hour sensible heat and 7000 Btu per hour latent heat. With 80 F maintained in the rooms, the restriction of 70 F for the temperature of the air leaving the registers, together with the known rise of 4 F in the duct system, determined that the temperature drop of the air passing through the coil should be 15 F; or from 81 F to 66 F. This drop in temperature, together with the sensible heat load, limited the air requirement to 1300 cfm. A net face velocity of 365 fpm permitted the use of an air duct of reasonable dimensions, and thus fixed the face area of the coil. The freedom of choice then proved to be in the amount of coil surface, or the number of rows of coil to be used.

The curves in Fig. 8 were computed from the data furnished by the manufacturer of the coils, and indicate that the required temperature drop in the air could be obtained either by the use of a large amount of cooling water with a small amount of coil surface, or by a small amount of water in connection with a large amount of coil surface. Furthermore, the resistance to the flow of air increased rapidly as the amount of surface, or number of rows of coil increased. In addition to the possible limit imposed on the air delivery by the higher resistance, the use of a large amount of surface has the further disadvantage of high first cost. On the other hand, the use of a small amount of coil surface has the disadvantage of high operating cost due to the amount of water used. Under these conditions a compromise is necessary. In the case of the Research Residence, this was effected by selecting 8 rows of coils. As shown by the water rate curve in Fig. 8, this determined a water rate such that a decrease of one or two rows would have resulted in an appreciable increase in water consumption, while an increase of one or two rows would not have resulted in an appreciable decrease in water consumption.

The resistance offered by the coil itself to the flow of water was only 0.5 lb per square inch as shown in Item 20 of Table 1. The drop in pressure from the city main to the entrance of the coil, however, was approximately 35 lb per square inch through the ¾ in. line from the main to the coil. This loss in pressure may in some cases be the limiting factor in determining the amount of water available for cooling purposes. The estimated resistance to flow of air of the 8 row coil was 0.29 in. of water.

Under normal conditions of winter operation, the total resistance of the system, including friction and shock losses, was 0.18 in. of water, and the fan as installed delivered 1675 cu ft of air per minute. For the summer work, however, in order to facilitate testing, particularly in connection with the mea-

surement of the quantity of air delivered, the cooling coil was located in a by-pass in the main return air duct, and the two auxiliary returns were blocked, thus causing all of the return air to pass through the coil and the single main return duct. In addition, the delivery ducts to the sun room and third story were blocked, and it was necessary to readjust the dampers in the remaining

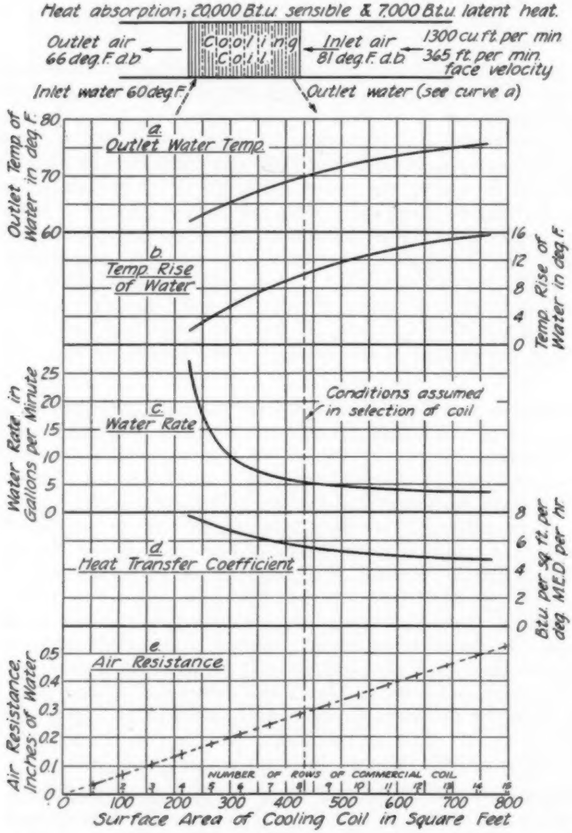

FIG. 8. COMPUTED PERFORMANCE CHARACTERISTICS FOR ONE TYPE OF COOLING COIL

delivery ducts in order to balance the cooling on the first and second stories. These changes had the effect of increasing the total resistance of the system without the coil to a value of 0.28 in. of water with 1347 cu ft of air flowing per minute. As shown in Item 24, Table 1, the measured resistance of the dry coil with this amount of air flowing was 0.23 in. of water. The latter was greater than the total resistance of the system during winter operation, and

was comparable with that of the system without the coil in the case of summer operation. The total resistance of the system with the coil in place was 0.51 in. of water. Under these conditions it was found necessary to install a larger fan for the summer work.

When the surface of the coil became wet with water condensed from the air, the measured resistance of the coil alone increased from 0.23 in. of water to 0.28 in. of water; or 22 per cent. With the speed of the fan maintained constant, the increase in resistance resulting from the collection of moisture on the coil surfaces was sufficient to cause variations in the amount of air flowing, depending on the amount of surface wetted. The amount of air delivered through the dry coil was 1347 cfm, and with maximum condensation occurring it was 1277 cfm; or a reduction of 5.2 per cent.

If the cooling coil had been located in the return duct just above the fan inlet without further changes in the winter system, it is possible that the total resistance would have been increased to only 0.41 in. of water instead of to 0.51 in. of water. Even this, however, is more than double the normal resistance for winter operation, and it is doubtful whether the fan used for winter operation could have been retained. In cases where the temperature of the available service water is above 55 F, the resistance of the coils and the possibility of the need for a larger fan for summer cooling than for winter heating should be given careful consideration in the design of forced-air systems that are to be used for both summer cooling and winter heating.

Summary of Seasonal Results

Table 3 gives a summary of the total quantities obtained for the season of 1935. The total of 1081.8 degree-hours above 85 F indicates that this season was mild as compared with those for 1934, 1933, and 1932, for which the degree-hours above 85 F were 2657.5, 2309.3 and 1470.7 respectively. This was caused by the comparatively small number of days on which the outdoor temperature rose above 90 F. The hours above 85 F for this season were 322.6 as compared with 481.5, 493.2 and 329.1 for 1934, 1933 and 1932, while the hours above 90 F were only 82.5 as compared with 224.1, 208.4 and 121.8 respectively for the previous seasons. The severity of the season as measured by degree-hours is apparently determined largely by the total number of days on which the maximum temperature exceeds 90 F, since the higher numbers of degree-hours are always accompanied by the higher numbers of hours above 90 F. While the 1935 season was mild from the standpoint of outdoor temperatures, it was characterized by unusually high outdoor humidity. This is indicated to a certain extent by Fig. 6, from which it may be observed that the outdoor specific humidity ranged from 92 to 140 grains per pound of dry air in 1935 as compared with a range of from 60 to 118 grains per pound for 1934.

From Item 6, Table 3 it may be observed that the total time that the fan was running during periods when it was circulating air from outdoors at night was 706 hours. At the local rate for electrical current of 3.1 cents per kilowatt-hour prevailing in Urbana, Ill., the cost of operating the fan during these periods was 1.37 cents per hour. During the stand-by periods from 7:00 a.m. until the indoor temperature rose to 80 F the fan was allowed to recirculate the air in the house. As shown by Item 10 the total running time for the fan during these periods was 474.8 hours. During all of the time that cooling was

demanded the fan was allowed to run irrespective of whether the water control valve was open or closed. The total time that the fan was running during these periods, as shown by Item 14, was 201.1 hours. The total time during which the fan was operated to recirculate the air in the house was therefore 675.9 hours, and the cost of operating the fan to recirculate the air in the house was 1.31 cents per hour. The total amount of water actually used during the total running time of 120.6 hours was 43,300 gal. At the prevailing rate of

TABLE 3. SUMMARY OF RESULTS OF TESTS USING WATER FOR COOLING AND USING NIGHT AIR COOLING FOR ENTIRE SEASON OF 1935

Weather Data	
1. Total hours above 85 F for season of 1935	322.6
2. Total hours above 90 F for season of 1935	82.5
3. Total degree-hours above 85 F for season of 1935	1,081.8
4. Total degree-hours above 90 F for season of 1935	174.1
Night Air Cooling	
5. Total number of nights with night air cooling by fan	63
6. Total running time for fan during night air cooling, hours	706
7. Average rate of power input to fan during night air cooling, watts	443
8. Total power input to fan during night air cooling, kwhr	313.1
Air Recirculation without Water Cooling	
9. Total number of days with recirculation of house air by fan	59
10. Total running time for fan during recirculation of house air, hours	474.8
11. Average rate of power input to fan during recirculation without water cooling, watts	415
12. Total power input to fan during recirculation of house air without cooling, kwhr	196.9
Air Recirculation with Water Cooling	
13. Total number of tests with water cooling	32
14. Total running time for fan during test period, hours	201.1
15. Average rate of power input to fan during test period, watts	433
16. Total power input to fan during test period, kwhr	87.1
Combined Totals	
17. Total running time for fan including night air cooling, hours	1,381.8
18. Total power input to fan including night air cooling, kwhr	597.1
19. Total number of hours of test period for season	201.1
20. Total number of hours of water circulation during test period	120.6
21. Total quality of cooling water, gallons	43,000
cubic feet	5,690

33 cents per thousand gallons this represented an hourly cost of 11.8 cents per hour for water alone, or a total hourly cost of 13.1 cents for both water and electricity during the actual time when the water was running.

Owing to wide variations in seasonal demands and in local water and electrical rates, seasonal costs are comparatively meaningless. With restriction of due appreciation of their limitations, however, they are undoubtedly of some interest. During the part of the season over which the plant was in operation, the total cost for electricity for operating the fan both to circulate air from outdoors at night and to recirculate the air in the house during the day was $18.52. The total cost for water was $14.29, and the cost for both electricity

and water was $32.81. Although the cooling plant was not ready to operate before July 11, observations on the temperatures and conditions in the Residence were made dating from June 1. The first part of the summer, particularly during May, was unusually cool, and these observations indicated that between May 1 and July 11 only 8 days occurred on which any cooling whatever would have been required. If these days are included, a reasonable estimate of the cost of cooling for the season of 1081.8 degree-hours would be approximately $36.00.

CONCLUSIONS

The following conclusions may be drawn as applying to the Research Residence and the conditions under which the tests were conducted.

(1) Using water from the city mains at 58 F, it is possible to maintain an indoor dry-bulb temperature of 80 F with the outdoor temperature as high as 100 F. On days when the maximum outdoor temperature exceeds 90 F the plant will operate continuously over considerable periods of time and it is possible to maintain an indoor relative humidity of 63 per cent, with a resultant effective temperature of 75 F. While not representative of optimum comfort, these conditions do represent a material improvement in atmospheric conditions that would probably be acceptable to the majority of users.

(2) On days when the maximum outdoor temperature does not exceed 90 F the plant will operate intermittently at all times and the indoor relative humidity will exceed 63 per cent. Under these conditions the maintenance of an indoor dry-bulb temperature of 80 F will result in effective temperatures exceeding 75 F and conditions will not be satisfactory from the standpoint of comfort.

(3) On moderately warm days it is possible to operate the plant with 58 F water from the city mains so as to maintain effective temperatures not exceeding 74 F with indoor relative humidities ranging from 70 to 80 per cent, and to maintain acceptable conditions from the standpoint of comfort.

(4) With 58 F water from the city mains it is possible to obtain sufficient dehumidification to maintain 63 per cent relative humidity indoors. These results indicate that 60 F is probably the upper practical limit for the temperature of the available cooling water unless an excessive amount of coil surface is used, in order to maintain indoor dry-bulb temperatures as low as 75 F in severe weather.

(5) In designing a plant to operate on 58 F water from the city mains the resistance of the coils to the flow of air and the capacity of the fan available may become limiting factors in determining the amount of surface and character of the coil to be used.

(6) The resistance of the coil to the flow of water is very small, but the pressure loss in the line between the coil and the city main may be large enough to become a limiting factor in determining the amount of water that can be circulated through the coil.

ACKNOWLEDGMENTS

The results presented in this paper were obtained in connection with the summer cooling investigation (1935) in the Research Residence,[8] Fig. 1, at

[8] The Research Residence in Urbana, Ill., was built, furnished, and completely equipped specifically for research work in warm air heating by the *National Warm Air Heating and Air Conditioning Association* in December, 1924.

the University of Illinois, conducted by the Engineering Experiment Station of which M. L. Enger, Dean of the College of Engineering, is the director, in the Department of Mechanical Engineering, of which O. A. Leutwiler, Professor of Mechanical Engineering Design, is the head. These results will ultimately comprise part of a bulletin of the Engineering Experiment Station. Acknowledgment is also due to J. S. Cunningham, Research Assistant, for services rendered in connection with the investigation, and to the various manufacturers who cooperated by furnishing instruments and equipment.

No. 1035

COMFORT STANDARDS FOR SUMMER AIR CONDITIONING

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This paper is the result of research sponsored by the American Society of Heating and Ventilating Engineers in its Research Laboratory at the Pittsburgh Experiment Station of the U. S. Bureau of Mines

The Effective Temperature Scale presented[1] to the American Society of Heating and Ventilating Engineers by its Research Laboratory in 1923, and the subsequent development[2] of the winter and summer Comfort Zones were assumed at the time to apply universally to indoor air conditioning, regardless of daily variations in the outdoor temperature and the length of exposure of occupants to indoor conditions. In applying these findings to air conditioning in theaters and elsewhere, it soon appeared, however, that the information available was not adequate and that some consideration should be given to the temperature of the outside air. It was assumed as a result of practical applications in the air conditioning field that rigid limits had to be applied to the moisture content of the indoor air in order to avoid sensible perspiration and other undesirable effects. Consequently there has been developed, largely through box office complaints in theaters, a variable indoor summer cooling standard based upon the outdoor dry-bulb temperature, which has been used as a standard in the air conditioning chapter of The A.S.H.V.E. Guide since 1933. This variable standard is presented in Table 1,[3] and the range in allowable indoor cooling conditions for an outside dry-bulb temperature range from 72.8 F to 80 F is plotted, *A-A*, on the psychrometric chart, Fig. 2.

It will be noted that this standard allows only a single definite moisture content or dewpoint temperature of the air, regardless of outside conditions, and

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[1] Determining Lines of Equal Comfort, by F. C. Houghten and C. P. Yagloglou, A. S. H. V. E. Transactions, Vol. 29, 1923, p. 163.
See also—Effective Temperature with Clothing, by C. P. Yagloglou and W. E. Miller, A. S. H. V. E. Transactions, Vol. 31, 1925, p. 89.
[2] Determination of the Comfort Zone with Further Verification of Effective Temperatures Within This Zone, by F. C. Houghten and C. P. Yagloglou, A. S. H. V. E. Transactions, Vol. 29, 1923, p. 361.
See also—The Summer Comfort Zone: Climate and Clothing, by C. P. Yagloglou and Philip Drinker, A. S. H. V. E. Transactions, Vol. 35, 1929, p. 269.
[3] A. S. H. V. E. Guide, 1935, Chapter 2, Table 2, p. 48.

Presented at the 42nd Annual Meeting of the American Society of Heating and Ventilating Engineers, Chicago, Ill., January, 1936, by F. C. Houghten.

a definite indoor dry-bulb temperature for any given dry-bulb temperature outside. Attempts to apply this standard in practice for different air conditioning applications and with different types of equipment have resulted in considerable chaos and dissatisfaction, resulting in the resubmission of the study to the Society's Research Laboratory.

In making the study, five students of the University of Pittsburgh acting as subjects judged their feelings of warmth in various atmospheric conditions in the psychrometric rooms of the Research Laboratory, located in the Pittsburgh Experiment Station of the United States Bureau of Mines. While the results

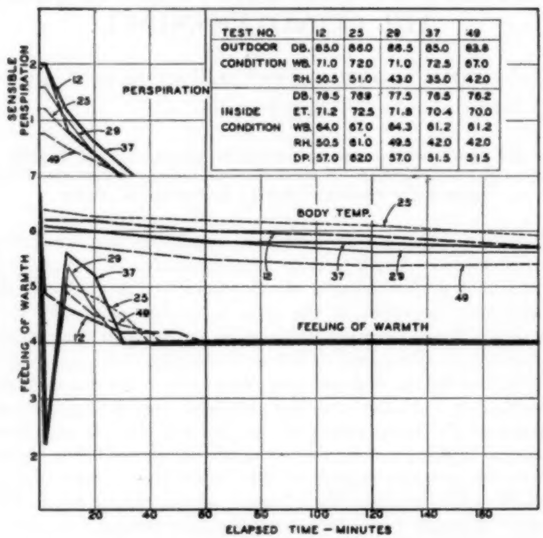

FIG. 1. AVERAGE REACTIONS OF FIVE SUBJECTS AFTER ENTERING VARIOUS COOLED CONDITIONS WITH MODERATE RELATIVE HUMIDITY

of the study are not entirely conclusive as regards all phases of the subject, they are particularly conclusive in demonstrating that the effective temperature of the indoor condition is of much greater importance than the moisture content of the air inside, or the dry-bulb temperature of the atmosphere outside. It is shown that air conditions ranging from 70 or 71 deg to 74 or 75 deg effective temperature give a feeling of comfort, as far as warmth is concerned, over a wide range of moisture content throughout the summer months, regardless of the prevailing outside temperature on any particular day.

TEST ARRANGEMENTS

One of the psychrometric rooms of the Laboratory, fully described in an

earlier report,[4] was arranged so as to accommodate five subjects without crowding and without their being subjected to undesirable variations in temperature, moisture content or motion of the air. Any condition desired for summer cooling between 30 and 90 per cent relative humidity could be maintained with ease. Relative humidities as low as 20 per cent could be had with greater difficulty, when it was necessary to reduce the occupancy of the room to two subjects.

Five male students between the ages of 19 and 23 years served as subjects. Examination showed them to be normal as regards their body temperature, pulse rate, respiration and other physiological reactions which might affect their temperature regulation under various atmospheric conditions. The clothing worn consisted of athletic underwear, light-weight shirt with collar attached, tie, light socks, low shoes, light-weight summer coat, and light flannel trousers.

TABLE 1. DESIRABLE INDOOR AIR CONDITIONS IN SUMMER CORRESPONDING TO OUTDOOR TEMPERATURES[a]

(*Applicable to Exposures Less Than 3 Hours*)

OUTDOOR TEMP. (DEGREE F) DRY-BULB	INDOOR AIR CONDITIONS WITH DEWPOINT CONSTANT AT 57 F		
	DRY-BULB	WET-BULB	EFFECTIVE TEMP.
95	80.0	65.0	73
90	78.0	64.5	72
85	76.5	64.0	71
80	75.0	63.5	70
75	73.5	63.0	69
70	72.0	62.5	68

[a] Loc. Cit. See Note 3.

Prior to most of the tests the subjects were exposed to prevailing outside air conditions without being shaded from the sun for a period of one-half hour or more before entering the test chamber. During the latter part of this period they walked ¼ of a mile and immediately before entering they individually recorded their feeling of warmth and the degree of perspiration on their face and body. Their feeling of warmth was graded according to the following arbitrary numerical scale: (1) cold; (2) too cool for comfort; (3) comfortably cool (not particularly uncomfortable, but a condition such that if the subject had a choice he would choose a somewhat warmer condition); (4) ideally comfortable as far as the feeling of warmth is concerned; (5) comfortably warm (not particularly uncomfortable, but a condition such that if the subject had a choice he would choose a slightly lower temperature); (6) too warm for comfort; (7) too hot. Perspiration was graded according to the following arbitrary numerical scale: (0) forehead or body dry; (1) forehead or body clammy (moist but not covered with visible perspiration); (2) forehead or body damp (perspiration just visible); (3) forehead or body wet (sweat covering the surface or standing in beads or drops); (4) perspiration on the

[4] Loc. Cit. See Note 1.

forehead runs down or perspiration on the body runs or wets through the clothing.

Immediately upon entering the psychrometric chamber, and at frequent intervals thereafter, the subjects again recorded their individual feelings of warmth and the degree of perspiration found on their body. The subjects also observed and recorded their rectal temperature, pulse rate and any other observations concerning their feeling in or reactions to the atmospheric conditions. These observations were all made and recorded by each subject without discussing them with anyone else. One observer was engaged throughout the test in maintaining desired atmospheric conditions, making a record of same, and observing any other unusual circumstances pertaining to the test.

For most tests the atmospheric conditions were chosen beforehand and maintained constant for a half hour before the subjects entered in order to have the walls in equilibrium with the atmosphere. This condition was then maintained throughout the test, which usually lasted three hours. For several tests the subjects were brought into the psychrometric chamber in which a constant air condition was maintained for a sufficient length of time for them to become in equilibrium with it, after which the atmospheric condition was changed slowly and uniformly so as to follow a predetermined path on the psychrometric chart. In these tests the rate of dry-bulb temperature change was not greater than 1 deg in 20 min. When time permitted, the change in the atmospheric conditions was reversed and made to return to the starting point so as to complete a cycle when plotted on the psychrometric chart. In a few of the constant condition tests the subjects entered the cooled psychrometric chamber, not from the outside, but from a second room heated to a predetermined high temperature and humidity. This was done so that in several tests with widely different atmospheric conditions in the chamber the preliminary exposure to high temperature and the resulting physiological reactions would be the same.

TEST RESULTS

During the early part of the study a number of tests were made in constant atmospheric conditions designed to give comfort in accordance with the present GUIDE standards,[5] or with moisture content at some definite percentage of relative humidity above or below this standard. These tests proved that conditions dictated by the present GUIDE standard invariably gave comfort, but wide variations in relative humidity also resulted in comfort provided the same effective temperature was maintained. The results of five of these tests are plotted in Fig. 1. The feeling of warmth and degree of perspiration are plotted according to the scales given above. The body temperature is also given.

The observations made just prior to entering the test chamber are plotted in all cases on the zero x axis. It will be observed that invariably the subjects felt very warm before entering and were perspiring in varying degrees. Immediately upon entering the cooled chamber a cool shock was usually experienced. However, this shock was of short duration, so that recovery to a feeling of comfort or warmer than comfortable was recorded by 10 min after entering. Usually from 20 to 45 min were required for all subjects to reach a comfort feeling of (4) or comfortable. Sensible perspiration usually disappeared rapidly. In most instances a comfort feeling of (4) was experienced about the

[5] Loc. Cit. See Note 3.

same time that sensible perspiration disappeared. The body temperature upon entering was usually close to a degree above the generally accepted normal of 98.6 F as a result of the subjects walking in the high outside temperature. Upon entering the cooled room the body temperature invariably dropped slowly and in most cases reached normal in two or three hours.

Fig. 2 shows the course followed and the reactions experienced by the subjects during the tests in which the atmospheric conditions were changed. The atmospheric conditions in the test chamber at the time of entry, the range of

TEST NO.		59	47	50	52	53	54
OUTDOOR CONDITION	DB		79.2	83.0	73.6	75.0	
	WB		72.5	71.0	66.0	69.0	
	RH		73.0	56.0	68.0	75.0	
INSIDE CONDITION	DB	72.5	74.8	75.6	73.5	77.0	75.1
	ET	71.2	70.0	71.0	70.1	70.5	70.4
	WB	69.4	63.0	65.0	65.1	60.9	63.3
	RH	88.0	52.5	58.0	65.0	39.5	52.5
	DP	68.5	58.0	59.0	60.5	50.0	57.0

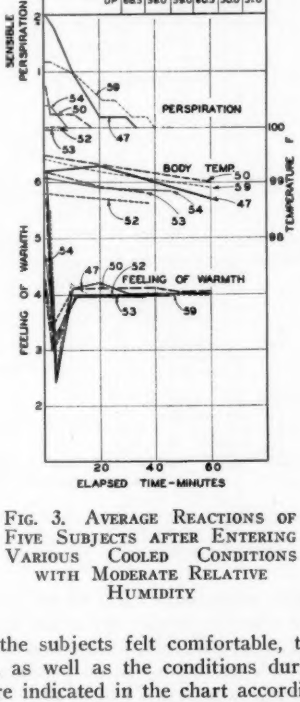

Fig. 3. Average Reactions of Five Subjects after Entering Various Cooled Conditions with Moderate Relative Humidity

conditions over which the subjects felt comfortable, too cool, decidedly cold, too warm and too hot, as well as the conditions during which they showed sensible perspiration, are indicated in the chart according to the key. During the first hour of these variable condition tests the atmospheric condition in the test chamber was kept constant so as to allow the subjects to become in equilibrium with it. The data obtained during this adjustment period for several tests in which all subjects become comfortable are plotted in Fig. 3.

After the approximate boundaries of the comfort zone were established by the variable air condition tests, Fig. 2, a few tests were made in various con-

stant air conditions in the apparent comfort zone or near its boundaries, in order to determine any variation in feeling upon entering such conditions from a hot atmosphere. Two series of such tests were made. In the first series tests were made in atmospheric conditions of 76, 73, 70 and 68 deg effective temperature and 85 per cent relative humidity, and tests at approximately the same effective temperatures but with relative humidity in the neighborhood of 30 to 40 per cent. All of these tests were entered from a hot condition of 86 F dry-bulb and 81.5 per cent relative humidity, maintained with small variation in an adjoining room. Another series of four tests was made in conditions

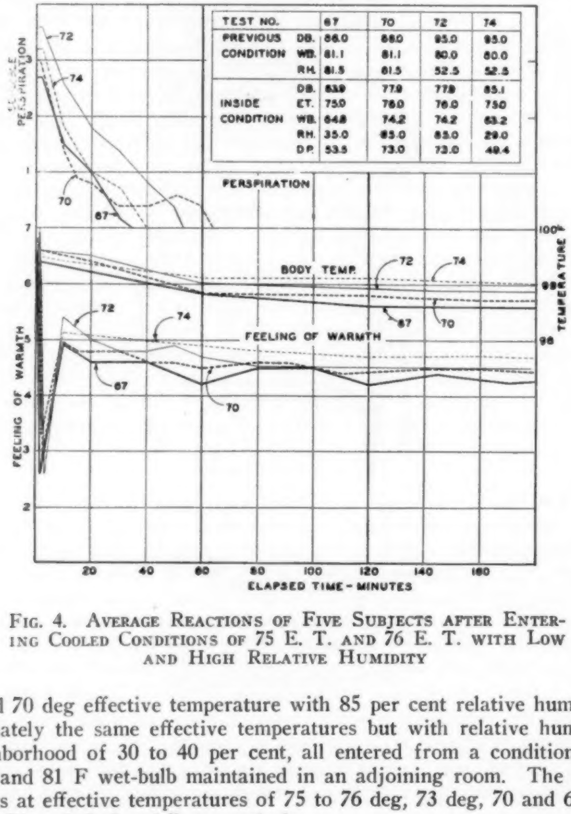

FIG. 4. AVERAGE REACTIONS OF FIVE SUBJECTS AFTER ENTERING COOLED CONDITIONS OF 75 E. T. AND 76 E. T. WITH LOW AND HIGH RELATIVE HUMIDITY

of 76 and 70 deg effective temperature with 85 per cent relative humidity, and approximately the same effective temperatures but with relative humidities in the neighborhood of 30 to 40 per cent, all entered from a condition of 95 F dry-bulb and 81 F wet-bulb maintained in an adjoining room. The results of these tests at effective temperatures of 75 to 76 deg, 73 deg, 70 and 68 deg are shown in Figs. 4, 5, 6 and 7, respectively.

The subjects showed mixed feeling of comfort and too warm or too cold in conditions near the upper or lower boundaries of the comfort zone. Tests made near the center of the zone usually showed entire agreement in the feeling of warmth of the five subjects as soon as equilibrium had been reached. In order to show these relationships the individual feelings of the five subjects

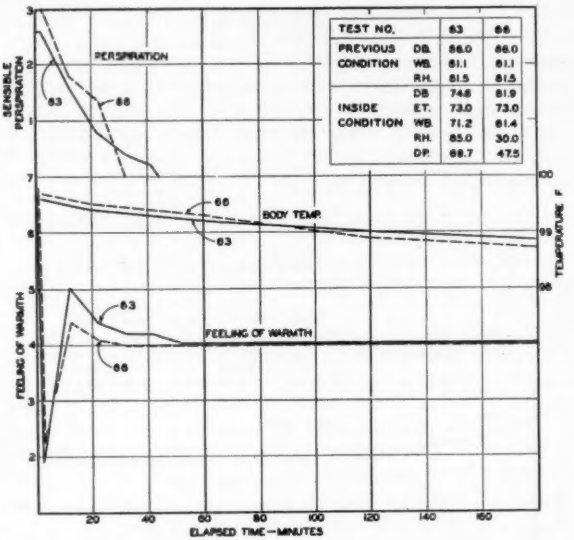

Fig. 5. Average Reactions of Five Subjects after Entering Cooled Conditions of 73 E. T. with High and Low Relative Humidity

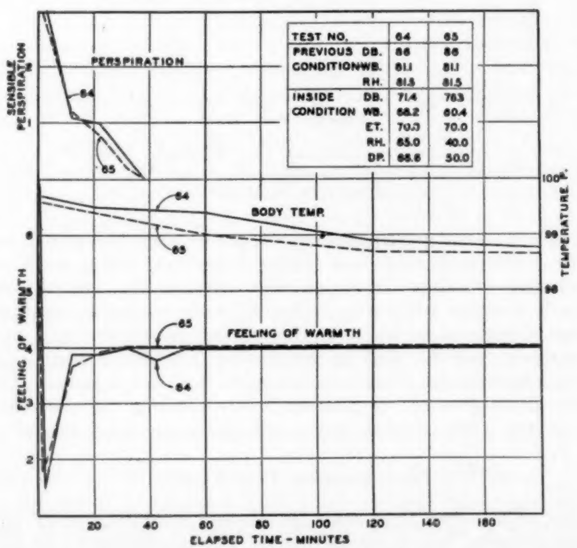

Fig. 6. Average Reactions of Five Subjects after Entering Cooled Conditions of 70 E. T. with High and Low Relative Humidity

in tests 69 and 73 are plotted in Figs. 8 and 9. The average values for all subjects in these two tests are plotted in Fig. 7.

A few pertinent facts concerning the 77 tests made during the summer are given in Table 2. The reactions of the subjects during the early part of the tests with changing air conditions are included. The tests are tabulated in the order of ascending effective temperatures of the air conditions. Besides the conditions of the atmosphere within the psychrometric chamber during the test, and the atmospheric condition from which the subjects entered, the table

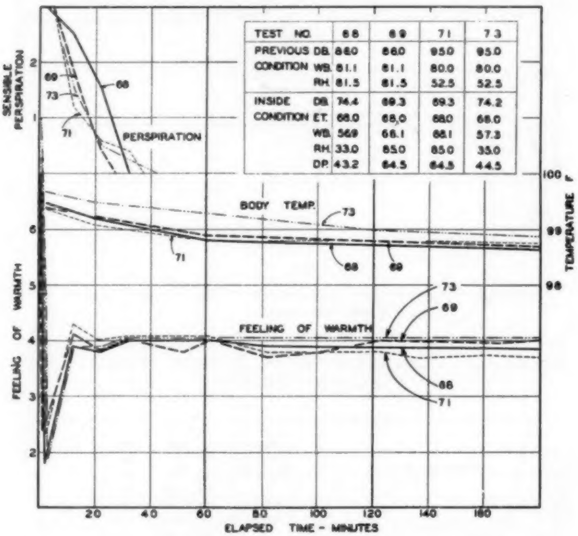

FIG. 7. AVERAGE REACTIONS OF FIVE SUBJECTS AFTER ENTERING COOLED CONDITIONS OF 68 E. T. WITH HIGH AND LOW RELATIVE HUMIDITY

gives the feeling of warmth experienced in preliminary condition before entering the air conditioned room, the lowest degree of feeling of shock immediately following entering, the elapsed time between the time of entering the psychrometric chamber and the time when all subjects became comfortable, and whether the subjects were warm or cool during the greater part of the time between entering and the time at which they reached comfort. Column O indicates whether the condition was warm or cool for comfort. Columns P and Q give the degree of perspiration before entering the test chamber and the lapse of time after entering before sensible perspiration disappeared.

DISCUSSION OF TEST RESULTS

Inspection of the four tests made in constant conditions at 75 and 76 deg effective temperature, Fig. 4, shows that the average feeling of all the subjects

never reached comfort for these effective temperatures either with low or high relative humidity. Actually, a mixed feeling of too warm and comfortable was indicated by the individual subjects in these tests. The two tests made at 85 per cent relative humidity show a greater persistence in the appearance of sensible perspiration than was the case in the low relative humidity condition. It should be noted, however, that the effective temperature was one degree higher in the 85 per cent relative humidity test.

Inspection of the variable condition tests, plotted in Fig. 2, shows a rather sharp division between reactions of comfort and too warm at about 75 deg effective temperature. Actually there is some overlapping as would be expected in data of this kind. Also there is a little tendency for the reactions felt by the subjects to lag, giving a feeling of comfort at a higher temperature when the air condition was changing to warmer, and lower when changing to lower temperatures. In no case, however, was a feeling of too warm expressed as much as one degree effective temperature below 75 deg effective temperature, or of comfort by as much as two degrees above. The varying condition tests, plotted in Fig. 2, indicate clearly that sensible perspiration appears at effective temperatures but little higher than those at which a feeling of too warm is experienced. These tests indicate a perspiration line somewhere between 76 and 78 deg effective temperature, with considerable likelihood of sensible perspiration appearing when the effective temperature is 76 deg or higher.

The constant condition tests, plotted in Fig. 1, and the variable condition tests, plotted in Fig. 2, indicate a considerable likelihood of a feeling of coolness when the effective temperature is below 70 deg. This boundary of the comfort zone is, however, not quite as sharply indicated by the tests as is the upper limit of 75 deg effective temperature. Some tests show all subjects to be comfortable at 68 deg effective temperature, while in some instances comfort was not reached even at a little above 70 deg effective temperature. There is a rather persistent tendency shown by the data for a feeling of comfort to be recorded at a lower effective temperature with low relative humidities than at high relative humidities. The study indicates that a feeling of comfort may be expected at an effective temperature 1 deg lower at about 30 or 40 per cent relative humidity than at 85 per cent relative humidity.

With very few exceptions, tests made in either constant or varying atmospheric conditions show that the subjects quickly experienced a feeling of comfort when the effective temperature ranged from 70 to 75 deg. In most tests made in conditions within this range, and more particularly within a range of effective temperatures from 71 to 74 deg, the subjects showed complete agreement in their feeling of comfort as regards warmth. As the limits of this range on either side are approached or passed, agreement was not so certain. This is brought out in Figs. 8 and 9, which give the individual feelings of warmth of the subjects in tests 69 and 73 at an effective temperature of 68 deg. The average results for the subjects in these tests are plotted in Fig. 7.

Discussion of Observations Made During the Study

Cool Shock on Passing from a Warm to a Cooled Atmosphere

The study shows a universal feeling of coolness upon entering a cooled atmosphere from a hot condition. Table 2 shows that this varied for the

different subjects in the different tests from a sensation of too cold or (1) to one of comfort or (4). On rare occasions no shock whatever was experienced.

TABLE 2. TEST DATA FROM STUDY OF SUMMER COOLING STANDARDS

A	B	C	D	E	F	G	H	I	J	K	L	M	N	O	P	Q
Test No.	Date	Time of Entry	Inside Test Conditions					Out Door Conditions			Degree Feeling of Warmth Before Entry-Test Room	Degree of Cold Shock Upon Entry-Test Room	Elapsed Time to Comfort in Minutes	Feeling During Majority of Time Before Comfort Was Reached	Deg. of Perspiration Before Entry-Test Room	Elapsed Time Until Sensible Perspiration Disappeared - Min.
			F ET	F DB	F WB	% RH	F DP	F DB	F WB	% RH						
48	8-5-35	10:10 A.M.	67.9	73.8	57.5	37.0	45.5	76.0	61.0	52.5	5.6	2.5	20	WARM	0.5	15
44	7-31-35	9:58 A.M.	68.0	73.5	58.5	41.0	48.0	75.0	70.4	80.0	5.0	2.6	**	COOL	1.0	15
40	7-29-35	10:29 A.M.	68.6	73.5	60.5	47.5	52.1	74.9	67.2	89.0	6.0	3.8	20	WARM	1.4	15
43	7-30-35	1:04 P.M.	68.7	75.0	58.0	35.0	45.5	80.0	62.2	36.5	5.9	3.2	30	WARM	0.5	6
24	7-17-35	10:06 A.M.	68.8	73.8	60.6	47.1	52.0	76.0	64.5	54.0	6.0	4.8	**	COOL	1.2	12
38	7-26-35	10:04 A.M.	68.9	74.0	60.6	46.5	52.0	76.8	71.2	77.5	5.8	3.2	10	COOL	1.2	18
23	7-15-35	10:01 A.M.	69.0	74.1	60.6	46.0	52.0	77.0	67.0	60.0	5.8	2.5	20	WARM	1.0	12
45	7-31-35	1:32 P.M.	69.2	76.0	58.0	38.5	45.0	83.5	74.0	65.0	5.5	1.5	60	COOL	0.7	10
26	7-18-35	10:00 A.M.	69.2	74.7	60.8	45.0	51.8	79.0	68.8	60.0	6.0	2.8	20	WARM	2.0	25
17	7-9-35	10:00 A.M.	69.4	73.5	63.0	57.0	57.0	74.7	68.0	71.0	6.0	2.5	**	COOL	2.0	10
32	7-25-35	10:10 A.M.	69.5	74.9	60.9	45.0	52.0	79.5	72.5	73.0	6.2	2.6	30	WARM	1.4	15
28	7-19-35	10:00 A.M.	69.5	75.0	60.9	44.0	52.0	80.0	71.5	67.0	6.4	2.6	20	WARM	1.8	25
11	7-2-35	10:04 A.M.	69.7	74.0	63.1	56.0	57.0	76.5	68.0	66.0	6.0	4.0	**	COOL	1.8	25
15	7-8-35	10:15 A.M.	69.8	74.2	63.2	55.0	57.0	77.0	70.0	71.0	5.7	3.8	**	COOL	0.8	25
6	6-27-35	10:54 A.M.	69.8	73.8	63.1	57.0	57.0	76.0	68.5	69.0	6.4	5.7	40	COOL	2.3	45
31	7-22-35	1:33 P.M.	69.8	75.7	61.0	43.0	51.8	82.5	72.2	61.0	5.6	2.8	20	WARM	0.6	15
36	7-25-35	10:00 A.M.	69.9	74.3	63.1	55.0	57.0	77.5	72.5	79.0	5.2	3.0	20	WARM	1.2	15
8	6-28-35	11:32 A.M.	70.0	74.4	63.5	55.5	57.0	80.0	71.0	65.0	7.0	4.0	**	COOL	2.5	40
49	8-5-35	1:34 P.M.	70.0	76.2	61.0	42.0	51.8	83.8	67.0	42.0	6.0	2.3	44	WARM	0.8	30
21	7-12-35	10:16 A.M.	70.0	76.0	61.0	42.0	51.0	83.8	71.0	54.0	6.0	3.0	45	COOL	2.0	25
13	7-3-35	10:10 A.M.	70.0	74.6	63.2	55.0	57.0	78.0	69.0	64.0	6.0	3.0	40	COOL	2.1	30
9	7-1-35	11:04 A.M.	70.0	74.6	63.3	55.0	57.0	78.5	65.4	50.0	7.0	3.0	**	COOL	2.6	20
65	8-27-35	10:40 A.M.	70.0	76.3	60.4	40.0	50.0	86.0*	81.1	81.5	6.9	1.8	30	COOL	3.0	38
64	8-26-35	1:29 P.M.	70.0	71.4	68.2	85.0	66.6	86.0*	81.1	81.5	7.0	1.5	52	COOL	3.2	38
52	8-8-35	10:10 A.M.	70.1	73.5	65.1	65.0	65.5	73.6	68.0	68.0	4.3	3.4	10	COOL	0.0	00
18	7-10-35	1:29 P.M.	70.1	76.2	61.0	42.0	51.5	84.0	71.5	55.0	7.0	2.6	50	WARM	3.2	40
19	7-11-35	9:59 A.M.	70.2	75.0	63.5	53.5	57.0	80.0	70.0	61.5	6.0	2.5	30	WARM	1.0	25

Invariably where the shock was experienced it had disappeared within 10 min. Figs. 8 and 9 show that the individual reactions of the subjects during this period were quite variable. This shock was experienced by the subjects without discomfort. While the condition felt cool, the contrast with the hot condition

was pleasing rather than uncomfortable in most instances. In this connection it should be kept in mind, however, that the subjects were all young men with

19	7-11-35	9:59 A.M.	70.2	75.0	63.5	53.5	57.0	80.0	70.0	61.5	6.0	2.5	30	WARM	1.8	25
47	8-2-35	10:04 A.M.	70.0	74.8	63.0	52.5	56.0	79.2	72.5	73.0	6.2	2.4	30	WARM	2.0	33
37	7-25-35	1:35 A.M.	70.4	76.5	61.2	42.0	51.5	85.0	72.5	55.0	6.6	2.4	30	WARM	2.0	34
54	8-12-35	10:09 A.M.	70.4	75.1	63.3	52.5	57.0				4.8	3.3	12	COOL	0.4	10
53	8-9-35	10:25 A.M.	70.5	77.0	60.9	39.5	50.0	75.0	69.0	75.0	4.3	3.0	12	COOL	0.0	00
7	6-27-35	2:54 P.M.	70.5	75.3	63.6	53.0	57.0	81.0	69.0	55.0	6.4	4.6	**	COOL	1.8	15
27	7-18-35	1:35 P.M.	70.7	77.1	61.2	40.5	51.1	87.0	72.0	48.0	6.8	1.8	20	COOL	2.2	35
30	7-22-35	10:00 A.M.	70.7	74.1	66.0	67.0	62.5	76.8	72.0	79.2	4.6	2.4	**	COOL	0.4	8
34	7-24-35	10:04 A.M.	70.9	74.4	66.0	66.0	62.0	77.8	72.5	78.0	5.1	3.1	20	WARM	1.0	15
33	7-23-35	1:28 P.M.	71.0	74.7	66.0	65.0	61.8	79.0	73.4	77.5	5.4	2.8	30	WARM	1.0	12
10	7-1-35	2:30 P.M.	71.2	76.5	64.0	50.5	57.0	85.0	65.0	34.0	6.2	5.0	80	WARM	1.6	35
12	7-2-35	1:39 P.M.	71.2	76.5	64.0	50.5	57.0	85.0	71.0	50.5	5.6	4.9	55	WARM	2.0	30
16	7-8-35	1:44 P.M.	71.2	76.5	63.9	50.0	56.6	84.6	74.0	61.5	6.5	2.2	55	WARM	1.8	40
39	7-26-35	1:30 P.M.	71.2	76.5	64.1	51.0	57.0	83.0	73.0	63.0	6.0	3.4	55	WARM	1.4	25
14	7-3-35	1:45 P.M.	71.4	76.8	64.1	50.0	57.0	86.0	71.8	50.0	6.0	4.5	40	WARM	2.1	55
22	7-12-35	2:00 P.M.	71.6	78.5	62.0	39.5	52.0	91.0	73.0	43.0	6.5	3.0	40	COOL	1.5	40
20	7-11-35	1:34 P.M.	71.8	77.4	64.3	50.0	57.0	88.0	74.2	53.0	6.8	2.6	55	WARM	2.2	20
29	7-19-35	1:30 P.M.	71.8	77.5	64.3	49.5	57.0	88.5	71.0	43.0	6.6	2.2	30	WARM	1.6	30
41	7-29-35	1:30 P.M.	71.9	75.9	66.5	61.5	62.0	83.0	68.5	48.0	6.2	4.6	50	WARM	1.8	25
35	7-24-35	1:34 P.M.	72.4	76.8	67.0	61.0	62.0	85.0	74.4	61.0	5.4	2.4	20	COOL	1.0	15
25	7-17-35	1:30 P.M.	72.5	76.9	67.0	61.0	62.0	86.0	72.0	51.0	6.0	2.6	40	WARM	1.2	30
63	8-26-35	10:35 A.M.	73.0	74.6	71.2	85.0	69.7	86.0*	81.1	81.5	7.0	1.5	52	WARM	2.6	44
66	8-27-35	1:30 P.M.	73.0	81.9	61.4	30.0	47.5	86.0*	81.1	81.5	6.8	2.1	30	WARM	3.0	32
67	8-28-35	10:25 P.M.	75.0	83.9	64.8	35.0	53.5	86.0*	81.1	81.5	6.6	2.6	**	WARM	2.7	34
74	9-3-35	1:44 P.M.	75.0	85.1	63.2	29.0	49.4	95.0*	80.0	52.5	6.9	2.9	**	WARM	3.2	40
70	8-29-35	1:40 P.M.	76.0	77.9	74.2	85.0	73.0	86.0*	81.1	81.5	6.8	3.3	**	WARM	3.0	64
72	8-30-35	1:25 P.M.	76.0	77.9	74.2	85.0	73.0	95.0*	80.0	52.5	7.0	3.3	**	WARM	3.6	52
56	8-15-35	10:24 A.M.	76.2	84.0	68.2	45.0	60.0				5.0	5.5	**	WARM	1.0	70
58	8-19-35	10:05 A.M.	77.0	86.5	67.5	37.5	57.0				4.8	6.0	**	WARM	0.2	180

* EXPOSURE PREVIOUS TO ENTERING THE TEST CHAMBER WAS IN A SECOND CONDITIONED ROOM.

** FEELING OF COMFORT NEVER REACHED. COLUMN "O" INDICATES WHETHER TOO WARM OR TOO COOL.

strong vitality and that the contrast upon entering a cooled atmosphere might be quite different for an older person having a lower vitality. Neither the magnitude or the duration of the shock bears much relation to the atmospheric condition maintained or its contrast with the previous condition, but apparently

the feeling of shock depends upon the degree of sensible perspiration present upon entering.

The shock upon entering a cooled atmosphere from one of higher temperature is usually considered psychological rather than physical or physiological. It can, however, be easily explained on a purely physical basis. When an average sized person is in a high temperature atmosphere, say 95 F dry-bulb and 80 F wet-bulb, earlier laboratory studies [6] show that he will dissipate a

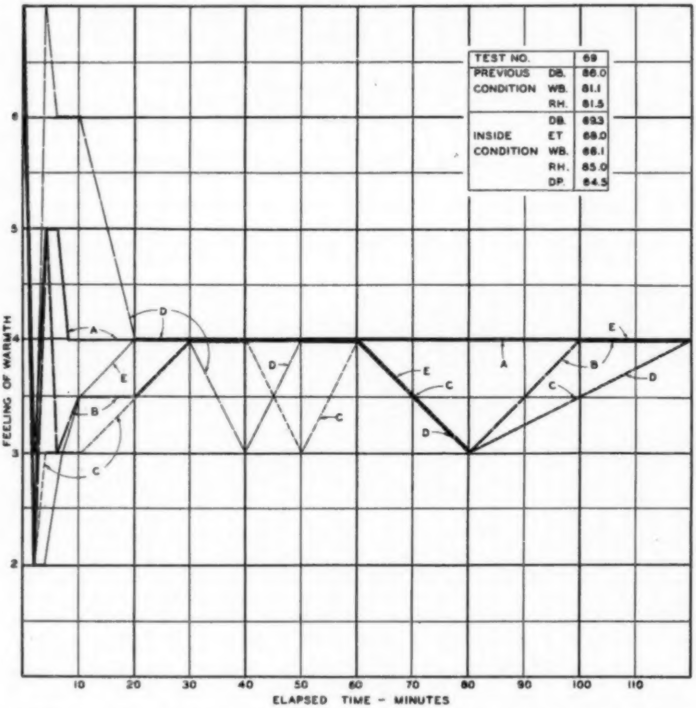

Fig. 8. Reactions of Five Subjects after Entering a Condition in Which They Were Slightly Cool for Comfort

total of 398 Btu of heat per hour, 52 Btu being eliminated as sensible heat by radiation and convection, and 346 Btu as latent heat of evaporation. In order to make this higher rate of evaporation possible, the body secretes perspiration at a sufficient rate to keep the body visibly wet. Under this condition evaporation is rapid; in fact, sufficiently rapid to take care of the 346 Btu of latent heat referred to. Sensible heat loss is reduced to 52 Btu by the small difference

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[6] Heat and Moisture Losses from the Human Body and Their Relation to Air Conditioning Problems, by F. C. Houghten, W. W. Teague, W. Ed. Miller and W. P. Yant, A. S. H. V. E. Transactions, Vol. 35, 1929, p. 245.

between the temperature of the surface of the body and the atmosphere. If this person suddenly passes into an atmospheric condition of 74 F dry-bulb and 64 F wet-bulb, the earlier laboratory findings [7] show that his sensible heat loss increases to 275 Btu per hour under normal conditions and latent heat decreases to 125 Btu per hour. Under the abnormal condition resulting from the sudden change, however, the body is wet with perspiration and the latent heat loss continues at the higher value 346 Btu per hour, making a total of latent

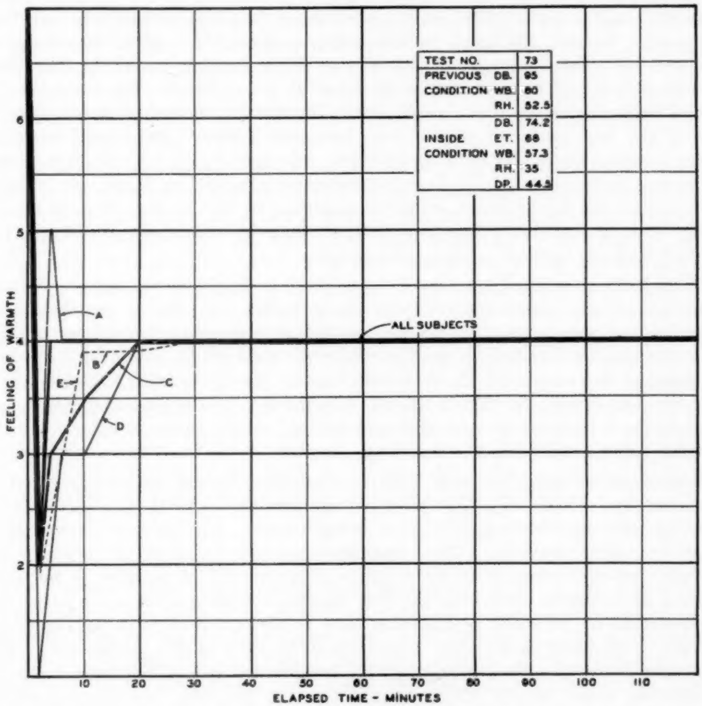


FIG. 9. REACTIONS OF FIVE SUBJECTS AFTER ENTERING A CONDITION IN WHICH THEY BECAME COMFORTABLE

and sensible heat loss upon entering the cooled atmosphere of 621 Btu, which naturally gives a sudden cool shock. It is natural, therefore, that the cool shock upon entering a cooled atmosphere from a hot one should depend more upon the condition of the body, including the perspiration thereon at the time of the change, than on any other factor. A study of skin temperatures during this transition would be very interesting and is recommended as a part of a continued study of this phase of the subject.

[7] Loc. Cit. See Note 6.

Relative Humidity Limitations

Considerable confusion exists regarding the effect of relative humidity on the comfort of occupants of space cooled during the summer. Sufficient time was not available during the summer to make a comprehensive study of small variations in reactions to small differences in relative humidity. This phase of the study should receive further attention by the Laboratory in a continued study of the entire subject. The observations which were possible during the study of feeling of warmth indicate rather conclusively, however, that relative humidity over a considerable range is of much less importance than had been supposed. In fact, the study indicates that a person having no knowledge of atmospheric conditioning in general and of the particular conditions pertaining around him seems to have little conception of the moisture content of the air. There was a tendency for those subjects in this classification to interpret as humid any hot condition where they perspired freely. Some such reactions were recorded when the relative humidity was actually as low as 30 per cent. A person more accustomed to air conditioning seems to judge the relative humidity of the air in which he is located more by the feeling of clothing and other objects outside of his body, rather than by sensations resulting from contact between his body and the atmosphere.

If high relative humidities are to be avoided it should be on account of their effect on objects within the occupied space, rather than due to any feeling of the occupants themselves. At 85 per cent relative humidity the feeling of paper and the way it handles is greatly affected, particularly when the dry-bulb approaches or exceeds 75 F. Colored drawing pencils used by the subjects in plotting data during the tests tended to become soft and unusable, carbon paper used in the typewriter became soft and smudgy, and attempts to erase on type-written sheets resulted in a smeared appearance.

Atmospheric conditions with relative humidities in the neighborhood of 20 per cent and around 75 deg effective temperature occasionally gave a dry burning sensation immediately after being entered, particularly when entered from a cooler condition. This sensation, however, quickly disappeared and probably resulted from lack of immediate physiological adjustment in the secretion of perspiration upon entering this condition from one where a lower rate of secretion of invisible perspiration was required. A person who does not perspire easily may have erratic sensations of warmth in these conditions. This sensation is never experienced when the conditioned space is entered from one sufficiently warm so that the body is perspiring sensibly. As a result of the study made during the past summer, it appears that no limitation need be placed on relative humidities between 30 and 60 per cent and that there is a possibility that further study may further expand these limits. Continued study of this phase of the subject is strongly recommended.

Sensations Upon Leaving a Cooled Atmosphere

A marked reaction is always experienced upon leaving a cooled atmosphere and re-entering the outside high temperature atmosphere. The experience of those taking part in the study during the past summer indicated that this reaction is of rather short duration. The sensation seems to be one of burning heat on the surface of the body, particularly where it comes in direct contact with the atmosphere. The men acting as subjects frequently observed a par-

ticularly burning sensation of the skin on the face, hands and on the lower part of the legs. This sensation disappeared after from 5 to 20 min. It probably results from a lag in establishing perspiration at a sufficient rate to take care of the higher rate of evaporative cooling required in the warmer atmosphere where sensible cooling is reduced.

Variations in the Indoor Cooling Standard with Outdoor Temperature

A great deal of attention has been given in recent years to variations in the dry-bulb temperature of indoor cooled conditions with variations in the daily outside temperature. The time available during the short summer period did not permit an intensive study of this phase of the subject. Such observations as were made did not, however, indicate any wide variation in comfort at any given effective temperature depending upon the outside temperature prior to the subjects entering the test chamber.

Probably of greater importance is the question of variation in the desired indoor conditions with the average daily maximum temperature throughout the summer season in any particular latitude. In this connection it is frequently assumed that because of the higher average daily maximum temperature in southern districts a higher effective temperature should be maintained indoors. Considering the fact that the comfort zone moves from a range of 63 to 70 deg effective temperature to a range of from 70 to 75 deg effective temperature for the latitude of Pittsburgh between the period of winter heating and summer cooling, it is natural to suppose that there will be a further change with more severe summer conditions. It should be observed from the data plotted in Fig. 2, however, that if the comfort zone moves much higher with more severe summer conditions the perspiration line must also move upward. Data are not available to indicate whether this is true or not. This phase of the subject should receive further study both in Pittsburgh and in some southern city where the average daily maximum summer temperature is higher.

Summary

The study indicates rather conclusively that the average person will become comfortable within from 20 to 40 min after entering any atmospheric condition within an effective temperature range of 70 or 71 deg to 74 or 75 deg. Within this range no appreciable variation was observed in the magnitude or the duration of the cooling shock with either the relative humidity of the cooled condition or the severity of the previous condition. Also, little variation in the rate of disappearance of sensible perspiration was observed for relative humidities up to 65 per cent; for relative humidities as high as 85 per cent there seems to be some increase in the length of time necessary for the complete disappearance of sensible perspiration.

DISCUSSION

Prior to the formal presentation of this paper at the Annual Meeting, several of the Society Chapters discussed this material at local meetings in which several of the members participated.

After the formal presentation of the paper by H. Harrison, the following com-

ments were presented at the Philadelphia Chapter meeting on November 14, 1935, at which J. H. Hucker acted as Chairman.

A. E. STACEY, JR.: The paper which has been presented for discussion is, in my opinion, one of the best pieces of research that we have had for several years. It is of particular interest to me because it continues the work in this field which was well started about 1921.

It appears to me that THE GUIDE table referred to in this paper may be set up as an operating standard rather than a design standard. In this particular table, inside dry-bulb temperatures are indicated as 72 F and it is my belief that most individuals have experienced that this value is too low. Generally, if the inside temperatures are below 74 F dry-bulb, there will be complaints from girls indicating that it is too cold. Perhaps, this value of 72 F was included in the table to satisfy the medical profession, as over a period of years health authorities have indicated that temperatures are not desirable for living conditions unless they approach limits from 70 to 72 F dry-bulb.

In connection with the upper limits of the values included in this table, I have had an opportunity to check these values over a period of years, in theatre application work. Ten years ago, we were particularly interested in the comfort of people in theatres and at that time, the inside dry-bulb temperature ranged from 78 to 80 F. Under these conditions of operation, we observed the wet-bulb temperature and watched the people in the room. If the wet-bulb increased it was noticeable that the occupants started to fan themselves and we attempted by this process of observation to note when the audience began to get uncomfortable. There was a large group of men taking these readings, and the result of these observations was that a wet-bulb temperature of 66 F and a dry-bulb temperature of 78 to 80 F seemed to be the most desirable. In connection with the paper presented, you will note that the high point on the curves was 80 F dry-bulb and 65 F wet-bulb which will indicate a point where people would start to fan themselves to obtain comfort in a theatre.

Some years ago Professors Yaglou and Drinker presented to the Society a paper on the comfort zone, applicable for summer conditions. In this investigation approximately 91 subjects were observed and their ages ranged from 16 to 70 years. By following the temperatures in the enclosure they subjected these people to the various conditions and then let them vote as to whether it was warm, too warm, etc., which is practically the same effort as has been used in this research. From these results an effective temperature of summer comfort was computed on a percentage basis. As I recall, with a 75 or 76 deg ET no one was comfortable and at about 71 deg ET, approximately 98 per cent of the people voted they were very comfortable, and at 66 deg ET everybody was too cool. Performing the tests in this manner permitted them to obtain results which would be comparable with crowds experienced by an engineer designing a store which would include people in numbers from 16 to 70 at least, and theatres where the occupants are all of varying ages. Hence, instead of using a limited number of five subjects, all of the same age and practically in the same condition of health, the experiments just referred to had subjects of every type. However, the surprising thing is that the results of the two investigations were extremely close. It is my recollection that in the experiments conducted by Professors Yaglou and Drinker, the optimum effective temperature was 71 deg, whereas, if the average of the results presented in this particular paper was assumed, the results would indicate only a variation of approximately 1½ deg ET above the chart submitted by Professor Yaglou.

This slight differential might be explained on a basis of the clothing worn by the various individuals and even from the particular location in which the people live. That is, the work done under Professor Yaglou's direction was accomplished at

Cambridge, Mass., while the work referred to in this paper was accomplished in Pittsburgh, Pa. Generally, people in Cambridge wear more clothes than they do in such territories as Philadelphia, because it is cooler.

It is my understanding that one of the important points in connection with this particular research is to determine the relative humidity limits, not only from a comfort standpoint, but also from the consideration of odors which may arise from furniture and other hygroscopic materials in the room. It seems to be a well established fact, that a room will have a peculiar odor if the humidity gets abnormally high. It may be that the 30 to 60 per cent relative humidity which has been suggested in this paper will keep the odor intensities within the limits desired.

Another project on which I hope a great deal more work will be accomplished is upon the shock experienced by individuals entering an air conditioned space from outside conditions. It may be that using a group of people of widely varying ages and physical condition, this shock effect may be quite different than the results indicated in this paper.

It is my belief that this work is being directed by men who have had more experience in this field than will be found in any other part of the world, so that the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS are particularly to be congratulated in being able to have work of this kind presented to them. It appears to me that this type of work has brought the attention of our Society to the whole world more than anything else that has been accomplished in the research laboratories.

C. S. LEOPOLD: The significant thing about this paper to me is that it accentuates a fact most of us have overlooked in confirming the accuracy of the original work on effective temperatures in the comfort zone. There has been a great deal of criticism of this work by engineers and people engaged in all forms of air conditioning and by the medical profession. In analyzing these criticisms, it is apparent that the people were reading their own interpretations into the results instead of accepting the data as they were presented.

The original comfort zone indicated that, within the range of 30 to 70 per cent relative humidity, the effective temperatures were about the same. It must be realized that the effective temperatures were all subjective tests recorded and observed by people, and this was the only way of obtaining the results. Based upon the results reported in this paper, it will be noted that, if we disregard the effects of the first hour of occupancy, the original effective temperature charts were correct. It is true that they were originally made with the greatest of care and accuracy and that approximately 20 times the number of subjects were used in connection with the results reported in this investigation. Another interesting point in connection with this paper is the question of shock and the conditions which immediately follow the initial effect. On an air conditioning job where the period of occupancy exceeds an hour, it makes little difference what the outside temperature is in reference to the inside conditions. People become acclimated to certain desired temperatures. An excellent example of this condition is in a department store where a customer enters a first floor where he is most liable to complain of conditions. The sales person at the same time has been in the enclosure all day and is also complaining, but the latter complains that it is too warm, whereas the customer complains because it is too cold. In stores having basements, it is common practice to maintain a lower effective temperature and the result is generally complete satisfaction. The problem of shock is rarely noticed after the elimination of the original conditions at the point of entering an air conditioned space.

While designing a job in New Orleans, a few years ago, I was much surprised to find that it was the easiest job I had ever experienced instead of the hardest one. In the first place, in the summer the men in New Orleans are dressed in cottons

and a great many go without coats, so that their bodily clothing was comparable to that of the women. The entrance to the store being air conditioned was formed by an arcade in which there were located several showcases and as an experiment a small amount of conditioned air was introduced into this space. There were no doors opening to the exterior from this space, but it was reasonably enclosed so that the effective temperature in this area was a mean between the store and the street. On this particular job, a complaint was almost unknown and, in the interior of the space, an effective temperature was maintained similar to that which is used in the Philadelphia area.

The interesting point of this whole study in my opinion is that shock period referred to which takes place in the first 30 min. As you will recall, the complete recovery from shock was about 30 min and the complete recovery from perspiration was of the same time duration. The recovery from the original shock was considerably less and seemed to coincide with reduction from perceptible perspiration to no perspiration observed on the forehead. Another interesting point referred to by Mr. Stacey was with reference to the odor conditions resulting from relative humidity conditions approaching 60 per cent. I think it is common experience, that in any place where there is tapestry, upholstery or hygroscopic materials, very few objections are experienced in connection with the presence of odors. In most places the relative humidity is liable to increase at night and would require lower relative humidities in the daytime to maintain levels for the prevention of odors. In an ordinary comfort installation using refrigeration with no supplementary means of reheating, it is my experience that we need not be concerned with relative humidities less than 45 per cent in our present methods of design.

These experiments are extremely interesting from the standpoint of confirmation, because the original experiments on comfort and effective temperature do not need confirming at this time, but rather they bring forcibly to our attention that after the first adjustment period the original laboratory results were very accurate and are still very accurate.

M. G. KERSHAW: In connection with an air conditioning system installed in a trust department of a bank, I should like to refer to some actual temperatures which were prevalent on one particular day when the outside temperature was 95 F dry-bulb, the inside conditions were maintained at 76.5 F dry-bulb, with a 57 per cent relative humidity which was equivalent to a 71 deg ET. The same conditions were maintained with little variation in the coupon room located in the basement. One of the bank's important customers was a woman about 65 who came in to clip her coupons and she complained of chilliness under these conditions.

On the upper floors of the same building, temperatures were maintained based upon the desires of the various occupants for a period of about 3 weeks. Under these operating conditions, the building engineer had a telephone call on the average of every 5 or 10 min from the various occupants who complained of undesirable conditions. Of course, it is evident to engineers that temperature control will not stabilize immediately, yet this engineer was attempting to accommodate people and at the same time trying to hold his own particular position. Finally, the situation was solved by instructing the superior officer in that particular department to tell the engineer when to change the indoor conditions. Under this plan, the man entered the room in the morning and if he felt warm the conditions were set accordingly. On the other hand, if he came in and did not feel quite so peppy, the conditions were adjusted differently for this particular air conditioning system.

In the Board of Directors room, in which two of the members defied engineering knowledge by saying that a room that size could not be ventilated without drafts, we checked some 3500 kata thermometer readings to determine air motion. In addition, we floated balloons, used smoke bombs and every conceivable method of air

flow determination and we finally tested the results by placing 34 candles on the Board of Directors' table. In the final test, 34 individuals were seated at the table which included the Board of Directors, the construction superintendent, the contractor and ourselves, and it was interesting to note that the 34 candles burned vertically upwards. This is one installation in which no complaints were received except on a day when the outside conditions were 98 F dry-bulb and inside about 78 F dry-bulb. The Board of Directors is composed of young men, full of pep, and some of them approaching old age, and on this particular day the group stepped out into the corridor of the banking space about 4:30 in the afternoon and a noticeable effect of shock was experienced. In order to eliminate this condition, we instructed the secretary of the Board to adjust the thermostat a couple of hours before leaving the room, in order that the temperature differentials would not be too great. On the day referred to in this discussion, the secretary had turned the key on the automatic control instrument in the wrong direction and, of course, the various individuals complained of the extreme shock experienced upon leaving the room.

To obviate this condition, an automatic control arrangement was installed so that the inside condition was adjusted as the outside temperatures increased and decreased. The resultant system of operation does not conform to the conditions outlined on the Society Comfort Chart, but we have experienced no complaints in connection with this method of operation. The control is set, so that the room dry-bulb is 75 F inside, when the conditions are 75 F outside. When the outside temperature rises to 80 F the inside conditions are maintained at 77.5 F. At 85 F out-of-doors, it is operated so as to establish conditions of 80 F inside. At 90 F outside, it is 82.5 F inside. As conditions approach 95 F outside, a condition of 85 F inside is maintained. No effort is made to establish uniform humidity conditions and it is allowed to vary on this particular installation over wide extremities.

In industrial air conditioning applications comfort is always considered in connection with the workers, but, primarily, the control of the product is the first requirement. In one room of a textile plant conditions of 75 F dry-bulb with a 60 per cent relative humidity and a corresponding 70 deg ET were maintained. The occupants during the daytime have no complaints at all. However, if one should come out of that room, a shock will be experienced depending upon the direction one travels to rooms of higher or lower temperature. The maximum effect occurs when the occupant travels directly to the outside condition.

In a textile plant located near Nashville, Tenn., the air is passed through an air washer to maintain desirable indoor air conditions. Temperatures were observed on a day when the outside conditions were 92 F dry-bulb and 76 F wet-bulb with a corresponding effective temperature of 82.5 deg. The room conditions were maintained at about 82.5 F dry-bulb with an 80 F wet-bulb and a 90 per cent relative humidity. Another reading taken in the same room at a different time indicated an 82.5 F dry-bulb and 80 per cent relative humidity. These two inside air conditions corresponded to effective temperatures of 80.8 and 79.7 deg. There were 15 men working in a plant who varied in occupation from a bottle washer to a control chemist. The men in charge of the work included the control superintendent and his assistant, and although we were not advocating indoor conditions of the temperatures indicated, these particular men stated that they were comfortable.

Personally, I did not walk into this particular room and had no personal reaction, but I should like to have C. R. Williams express his personal reaction upon entering this space.

C. R. Williams: On entering the room it was noticeably cooler than the outside, provided one did not attempt to do manual labor. As soon as you attempted to accomplish manual labor, it was explained by the mechanics who came in to repair a pipe or do some such work, that it was the most uncomfortable place in the plant.

However, by the men who were working in the space for 8 hours a day and doing nothing manually, it was reported as very comfortable.

H. H. MATHER: Recently we have conducted certain field investigations in an attempt to determine the reaction of the average person to certain air conditions. In one installation, we attempted to maintain indoor conditions as recommended in THE GUIDE for a period of approximately one month. Questionnaires were submitted to a group of people occupying this particular space for an average occupancy of 8 hours a day. Of the 47 people answering the questionnaire, 34 reported that conditions were satisfactory, 7 indicated that it was too cold and 6 reported that conditions were too warm. Hence, it will be noted that the majority agreed that the indoor conditions were satisfactory.

In connection with the observations we have made regarding indoor conditions, it has been our experience that the age of the occupant has considerable to do with the environmental conditions maintained.

For that portion of the commercial field in which air conditioning will be installed in the future, it seems that a large amount of the work will be applied to buildings where the occupancy period is for periods of a half-hour or less. In connection with this problem, we have attempted to obtain the reaction of people under varying conditions and in one case it was noted that with an outside temperature of 82 F dry-bulb and 77 F wet-bulb, inside conditions of 81 F dry-bulb and 67 F wet-bulb were maintained. The vast majority of the people entering this particular space and occupying it for a period of a half-hour or less were of the opinion that the conditions were slightly cold, although the inside conditions were near the upper limit of the summer comfort chart.

It would seem to me that further investigations should be made by the Society in an attempt to determine a compromise condition which would be satisfactory for employees who occupy an enclosed space over a long period of time and the customer who occupies it for a half-hour or less.

F. D. MENSING: Under some circumstances, it is desirable to take an expediency point of view but, unfortunately, there are many engineers who are faced with the proposition of designing equipment for people of various types. It is a difficult matter to design installations on a basis of unknowns. We have to have some measure or point to start with in design and we cannot have a flexible yardstick. When Professors Yaglou and Drinker conducted their experiments it was discovered that we knew very little about the reaction of the human being. At that time, it was decided to request cooperation from the medical profession in an effort to find some standard, some yardstick, because in the final analysis it was the individual for whom we had to provide desirable conditions. It is my understanding that the Society appealed to the *American Medical Association* in Philadelphia and this organization indicated that they had no means of assisting us. However, they suggested that we apply to the government and a doctor was assigned to do work along with our Society. I do not know what has happened to that particular doctor nor the movement which was started at that time.

This paper was somewhat of a disappointment to me because the investigators seemed to start where they left off years ago. Today, nobody seems to know what the human being really wants. What is comfortable to an individual today is not comfortable to him next year and as engineers, we are all selling human comfort.

Air conditioning got its first start in this country in the industrial field and if you bring an industrial problem to an engineer, be it conditioning paper or silk, he is fully familiar with the character of the material and the results which have to be obtained. When the comfort of a human individual is considered, the conditions must be something that fit this unknown person.

Let us analyze these unknowns. In the first place, half of the individuals in the world are females. One out of four does not know what comfort is, because for one-quarter of her life she has no idea as she is generally a sick person. You have got to face that, as in some industries it is a special problem.

In the telephone business where several operators are functioning, in front of a switchboard, and they try to operate that particular room to please the people that are working there, little is accomplished until a chief operator is placed in charge, whose word is law.

With material we have known values to consider, as is true in any type of operating room. These knowns are arbitrarily there, something the engineer can work on and in the telephone business, it is the chief operator. She is the engineer. She turns it on and shuts it off; she opens the window and closes it.

In the case of the individual, you have roughly, a wet-bulb thermometer. The moisture is being given off by that individual. However, it differs from the wet-bulb thermometer in this respect that inside of the thermometer, there is a coil which we can assume for argument is heated with electricity and is throwing off a certain amount of electricity. This individual is one of the most marvelous machines known. There are human beings in the United States or under the United States flag who have lived subject to temperatures of 120 F up, to 60 F down with a difference of perhaps 180 F. Soldiers of the United States army are subjected to conditions from Fort Yuma, Ariz., to Alaska.

The human being will adapt himself to almost any condition and survive. I have a friend who comes from Brazil and he has outlined conditions there which are vastly different from anything that we are familiar with in this country. He comes to Philadelphia. In Philadelphia consider our own range of dry-bulb temperatures which vary from —11 F to 106 F above and we survive. We have the ability to cover a multitude of conditions.

As I view the subject, what we need is not something of expediency to try and suit this one and suit that one and suit the other one, but a standard from which we can work and say "Gentlemen, this is what you should have and you can vary it as much as you want, one way or the other". Then I think the engineer is going to get somewhere, the salesman is going to get somewhere and the manufacturer will be able to develop equipment to satisfy the individual idiosyncrasies of people desiring comfort. In all our work we want to establish, in some way, a definite standard and work from this basis, a yardstick that is as fixed as the standard meter in Washington.

MR. LEOPOLD: Mr. Mensing, how would it be to take the thermostat off a wall and put it on a heating coil inside the human individual?

MR. MENSING: You have pretty nearly got it as the English have already done that. They have an ingenious apparatus of a ball with the heating element on the inside for measuring the heat relations from a human being. It seems to me that the English people are progressing in the right direction and, although it isn't the ideal, they seem to be accomplishing results.

MR. LEOPOLD: Unfortunately, they are trying to determine the same result that Professor Drinker is attempting to measure.

S. H. KNIGHT: In all this discussion, I have been wondering why we are trying to reach a standard over a range as small as a degree and a half on cooling, when on ordinary heating systems our temperatures fluctuate over a range of 10 or 15 deg to accommodate our individual desires. Based on these conditions, it seems undesirable to establish such a close line in connection with cooling installations.

It seems surprising to me that a paper of this type is submitted upon the observations of 5 individuals. The average of 5 men, whether they are alike or not alike,

in my opinion, is of no significance. I have been operating an air conditioning plant for approximately 6 years in a large department store where between 25,000 and 100,000 people visit the place daily. It is my opinion that if one wants to attempt to air condition an enclosure to suit individuals of this type, observers should visit establishments of this type and listen to the comments of the various individuals. Most of the people who visit this store are 95 per cent women. They are of all ages from athletic high school girls to old ladies 85 years old. Some visit the store with very few clothes on and some come in with heavy fur coats. Frequently, the customer that comes in dressed in a heavy fur coat is waited upon by a woman in a georgette waist. When we attempt to maintain an air condition of heating or cooling for that type of diversity and set it within a degree and a half, it seems to me that we are approaching merely a theoretical viewpoint.

It is my understanding that Strawbridge and Clothier was the third department store of any size in the United States to install air conditioning, with perhaps Filene's in Boston and Hudson's in Detroit having made installations of this kind previously. Both of these stores happen to be members of our research association, so that I had the freedom of access to their operating records, which were similar to the conditions prevalent in our own store. Prior to the initial operation of our systems, these records from the other stores were reviewed carefully.

I was always impressed with the old signs that were used in front of theatres, dating back about 10 years "70 deg inside" or "20 deg cooler inside", which were both wrong in my opinion as a fixed differential or a fixed temperature was not sound. A graduated differential of operation was adopted 4 years before it was referred to in the manual and this system has been operated on it ever since with as much success as can be attained in an application of this type. This method of operation is established on only one man's opinion which is based on six-tenths of the differential between outside temperature and 70 F inside.

I have previously referred to women entering the store in heavy fur coats and it will probably astonish you to know that our cooling load is just as great in the winter as it is in the summer. Some of the heaviest cooling loads observed occur at Christmas time. In 66 days that our store was open last winter during the months of November, December and January, in the first floor and basement where air conditioning is installed, we heated 5 days and cooled 37 and recirculated the rest of the time without either one. It is evident that our load is principally lighting and body heat.

The shock referred to in this discussion with reference to entering an inside condition from the outside is only half the problem as experienced by our observation. The shock that hits us right where it counts, and that is in the pocketbook, is when the individual leaves the air conditioned first floor to enter the second or third unconditioned floors where the principal amount of business is conducted. This is the reason we changed from a fixed differential of operation to a variable, as observations conducted in the Hudson store indicated astounding results. Customers shopping on the first floor with a cold condition ranging between 70 and 71 F on a hot summer day experienced noticeable effects when rising in an elevator to the third unconditioned floor. In one case I observed two little old ladies who experienced this change and when entering the third floor stated, "Get me out of here, I am going to faint. I can't stand this". Immediately, it was realized that if business was to be transacted on the unconditioned floors that the temperature on the lower floor must be warm enough so that the shock was not noticeable when leaving the cold condition and similarly, when entering such an environment.

The charts referred to in this discussion seem to be based upon stabilizing the body at a period extending over 3 hours. The average time of a person in our

store is approximately 32 min and under this condition, we do not experience the office building air conditioning problem.

Considering all things we must realize that air conditioning is not a matter of establishing a standard within a degree and a half or possibly 5 deg. It is really a matter of determining sound practice and then trying to fit the equipment to a particular job which has been engineered in accordance with our best judgment. If this is accomplished, I believe it is undesirable to attempt indoor regulations to even a degree and a half.

E. S. Bigelow: It seems to me from the charts presented in this discussion that the period of perspiration relief in almost every instance, was not less than 15 min and in a number of cases it extended to almost an hour. Our interest in selling air conditioning concerns the customer who has to pay for an installation out of the profits that are realized in its use and operation. Also, interest lies in those commercial establishments of the retail type where the greatest avenue of sales exists. A dress shop is an example of this type of installation and in these cases the period of occupancy for a customer is very short. Under these conditions, a woman comes into a shop where she remains for about 15 min trying on from one to two dresses during that time.

It is my impression that these experiments were conducted by observations of forehead perspiration. A woman sweats terrifically under the arms, particularly among big women. In our experiences, we have noted several complaints this summer from dress shop owners who have to send their garments out for dry cleaning because of this particular spoilage. Any possibility of this saving resulting from the application of air conditioning would in my opinion be of great value, if experiments could be conducted on women showing the perspiration period when reduced to a 5 or 10 min limit.

This year, observations have been relied upon solely by the owners of the equipment installed. Records have been maintained of the conditions established in some of the Philadelphia shops in order to inform prospective owners of the savings which they could expect from an installation of this type of equipment. It is interesting to note that there is a wide difference in temperatures and conditions given in these records as compared with those reported in this particular paper.

H. Harrison: Having presented the paper I had intended not to enter into this discussion, but there is one thing that strikes me forcefully in listening to these discussions. It seems to me that when the Research Laboratory conducts experiments, the men responsible for their direction and the preparation of the paper should present the material themselves and be here to listen to the various comments, because it is impossible to conceive of their getting the good out of the various comments by simply reading the discussion. It seems to me that the expression on the faces and the emotion expressed on the results is something that would be of invaluable help to the investigators. Another point that I would like to make is that progress is made in stages and in these tests irrespective of the fact that they are admittedly preliminary, there is one thing for which I feel I am at fault if you have missed it, and that is, one of the big objects of these tests was to try and indicate a trend. First, by showing that the standard in The Guide is not an infallible law and that any one, consulting engineer or operating engineer, who might insist that this standard be maintained is in error. An engineer might look in The Guide and say, "There it is, now I want that condition". For two reasons, he is in error. First, because the standard in The Guide does not necessarily mean that the condition is ideal for comfort and we have observed, from the discussions presented, the number of variables which need to be considered for establishing comfort conditions. The second, refers to the wide fluctuations in any given day of the

dry-bulb temperature. Any attempt which is made to try to maintain the relationship between dry-bulb temperature, outside and inside temperatures, as referred to in THE GUIDE is just impracticable. These two points, in my opinion, the authors of this paper have brought out in their test, by the conclusions that they have drawn and these particular things should not be overlooked in your analysis of this paper.

R. F. HUNGER: Is it true that the purpose of the authors was to present standards for a period of occupancy of 3 hours or more so that the comments on the 32 min occupancy would not apply to the recommendations made in this paper?

MR. HARRISON: The period of occupancy was as high as 2 hours and admittedly for from 20 to 45 min there was the experience of shock. The authors do not attempt in these tests, to give a solution to that part of the problem, if I interpret it correctly.

MR. HUNGER: In other words, the recommendations were not applied to a department store, but they might apply to a place like 12 South 12th St.

MR. HARRISON: They might.

J. D. CASSELL: As I understand the presentation of this paper at this time, it was to bring out any comments adverse or otherwise that could be presented in concrete form, when the paper is given at the January Meeting of the Society. It seems to me that we have had an interesting discussion here tonight and all of it is pertinent to the subject and when this record of discussion is presented at the Society Meeting, I feel sure it will indicate a very commendable work.

WESTERN MICHIGAN CHAPTER (J. H. VAN ALSBURG): This Chapter wishes to be recorded as approving the Comfort Standards for Summer Air Conditioning. It is realized that this project is tremendous and we commend the Society and its Research Laboratory for the fine work which has been done in the preparation of this and other papers on this same subject.

It is our belief that considerable more information on comfort standards is necessary and we hope that the Laboratory will be able to continue research along this line for some period of time, so that we will have adequate information. With reference to these standards, it is our suggestion that the following questions or suggestions be given further consideration: How was air introduced to the test room? At what velocity was the air introduced? What was the volume of the air handled? Where were the supply and exhaust openings located? What air movement was prevalent in the room introduced by the incoming air or caused by the proximity of the occupants? What was the approximate spacing of individuals who were subjects for these tests? What was the temperature differential between room air and admitted air?

According to the results reported in this paper, all of the subjects used were healthy young males which is hardly a cross-section of the average American. Inasmuch as the fair sex were not subjected to these experiments which may raise the effective temperature required to give desirable comfort, because of reduced amount of clothing or body insulation they usually wear, it is our opinion that elderly people should also be subjected to these tests.

The paper also refers to the psychrometric rooms in the Research Laboratory and reference is given to an early report which was published in 1923. It is our belief that a brief review of the equipment and the rooms used in these tests would be desirable because many new members are not familiar with the TRANSACTIONS dating back to 1923. We, therefore, suggest in this case, that an outline of the laboratory rooms be given mentioning the construction of the walls as well as the room size, etc.

It is assumed that this paper is not final and additional information will be written on this subject, which will have more practical upper and lower range values in the form of intermediate limits for effective temperature design.

MICHIGAN CHAPTER (A. C. WALLICH): At the December 16, 1935 meeting of the Chapter held at the Wardell Hotel, S. S. Sanford presented in an informal manner, the paper, Comfort Standards for Summer Air Conditioning. Mr. Sanford made a thorough study of the contents of the paper and prepared a number of slides to assist in his discussion. The paper provoked an outstanding amount of discussion and among the members participating were: Thomas Chester, J. H. Walker, G. B. Helmrich, C. L. Toonder, L. S. Keilholtz, E. M. Harrigan, A. H. Kirkpatrick, J. S. Kilner, W. A. Rowe, J. F. McIntire.

It was evident that our Chapter is enthusiastically in favor of the continuation of the subject of comfort conditions and comfort standards, and it was moved and supported and unanimously passed, that this Chapter advise the National Officers and the Committee on Research of our support for a further study of the conditions reported in this paper.

It was suggested that in any further studies which are made on this subject, that subjects of various ages and both sexes should be used to secure the reactions of average occupancy, simulating conditions in a theatre or crowded area. The statement was repeatedly made that young men of college age are far too robust and healthy to give us the average reaction of individuals in a cooled building.

C. TASKER (WRITTEN): May I compliment Mr. Houghten and his fellow workers on their presentation of the results of the experiments. Although the graphs look rather complicated at first sight, they are, when closely studied, quite simple, and express perhaps as clearly as one can ever hope to express graphically some rather complicated data.

I have been preparing some data in order to discover, if possible, what are the limitations surrounding the application to practical air conditioning problems of the comfort zone proposed as a result of this and other related studies.

TABLE A. METHOD OF COMPUTING THE MEAN EFFECTIVE TEMPERATURE BETWEEN 10 A.M. AND 9 P.M. (E. S. T.)

July 1926

Time of Day E. S. T.	Average of 31 Days Dry-Bulb, F	Average of 31 Days Rel. Hum., Per cent	Corresponding Effective Temperature
10 A.M.	71	62	68
11 A.M.	73	61	69
12 NOON	74	61	70
1 P.M.	75	58	71
2 P.M.	76	55	71
3 P.M.	76	54	71
4 P.M.	76	51	71
5 P.M.	75	51	70
6 P.M.	74	55	70
7 P.M.	71	60	68
8 P.M.	69	66	66
9 P.M.	66	72	64
Mean of 12 hours	73	59	69
Maximum E. T.	76	55	71
Minimum E. T.	66	72	64

TABLE B. AVERAGE EFFECTIVE TEMPERATURE BETWEEN 10 A.M. AND 9 P.M. (E. S. T.) AT TORONTO, ONTARIO

(Based on hourly meteorological data, with no allowance for wind)

	1923	1924	1925	1926	1927	MEAN OF 5 YEARS
June 21–30						
Mean Dry-bulb, F	75	75	66	69	74	72
Mean Rel. Humidity, Per cent	62	63	60	59	56	60
Mean E. T.	71	71	64	66	70	68
Maximum E. T.	74	74	65	68	72	71
Minimum E. T.	64	64	61	62	64	63
July						
Mean Dry-bulb, F	75	72	72	73	74	73
Mean Rel. Humidity, Per cent.	59	61	57	59	62	60
Mean E. T.	71	69	68	69	70	69
Maximum E. T.	72	71	71	71	72	71
Minimum E. T.	65	63	64	64	66	64
August						
Mean Dry-bulb, F	71	71	74	72	69	71
Mean Rel. Humidity, Per cent	58	64	62	73	62	64
Mean E. T.	67	68	70	70	66	68
Maximum E. T.	71	71	73	71	69	71
Minimum E. T.	62	63	65	66	63	64
September						
Mean Dry-bulb, F	64	60	65	62	67	64
Mean Rel. Humidity, Per cent	69	70	74	76	64	71
Mean E. T.	63	60	64	63	65	63
Maximum E. T.	65	62	66	63	68	65
Minimum E. T.	59	55	61	59	62	59

TABLE C. SUMMER CONDITIONS IN VARIOUS CANADIAN CITIES IN COMPARISON WITH THOSE OF PITTSBURGH

	PITTSBURGH	TORONTO	OTTAWA	WINDSOR	MONTREAL	WINNIPEG
Summer Mean Temp., F (June, July, and August 24 Hours)	73	66	66	69	67	64
Mean of Maxima, F	83	75	77	79	75	76
Mean of Minima, F	63	56	56	59	59	52
Humidity 8 A.M., Relative, Per cent	76	73	80	75[a]	77	..
Absolute, gr per cu ft	5.7	4.9	4.8	5.3[a]	5.1	..
Humidity 8 P.M., Relative, Per cent	62	72	77	64[a]	79	..
Absolute, gr per cu ft	5.8	5.1	5.3	5.4[a]	5.4	..
Total Sunshine, Average Hours per Month	277	268	253	301[a]	229	266

[a] Figures are for Detroit (U. S. Weather Bureau records).

The author states that the results of the study demonstrate conclusively that the effective temperature of the indoor condition is of much greater importance than the moisture content of the air inside, or dry-bulb temperature of the air outside.

It is also stated in the paper that, "Probably of greater importance is the question of variation in the desired indoor temperature conditions with the average daily maximum temperature throughout the summer season in any particular latitude", following this with the statement, that with more severe summer conditions than those experienced in Pittsburgh, a change in the comfort zone may be desirable.

Some of our biologists have been studying, during the past few years, the effect of environment on the growth and health of plants and animals, and have shown how important are such factors as fluctuations in dry-bulb temperature and relative humidity, length of day and hours of sunshine. With this in mind, and remembering how one gradually became acclimatized to Toronto summer conditions which, when first experienced on arrival from England, seemed almost oppressive, I have been wondering whether the data Mr. Houghten has obtained are not colored, shall we say, by the previous climatic history of the subjects who made the study. For instance, if into each of 3 psychrometric rooms maintained in the same condition as measured by the effective temperature, he placed 5 subjects normally resident in Toronto, Pittsburgh and New Orleans, how different would their reactions be?

I have been examining data published by the Dominion Meteorological Office in Toronto, and as a result have prepared several tables referred to in this discussion.

For many years, up to 1928, the hourly dry-bulb temperatures and relative humidities at several Canadian cities including Toronto were published by the Meteorological Office. Since it was thought that the 12 hours from 10 a.m. to 9 p.m., both inclusive, were perhaps of greatest interest in connection with the summer air conditioning of, say, movie theatres, the figures for Toronto were averaged for the periods June 21st to 30th, July 1st to 31st, August 1st to 31st, and September 1st to 30th. An example of the method of treatment is shown in Table A in which are given data for the month of July 1926.

The average dry-bulb temperature, at 2 p.m. was 76 F, the average relative humidity was 55 per cent and, *making no allowance for wind,* the effective temperature from the chart was 71 deg.

The mathematical mean of the 12 hours is shown at the bottom of the columns, i.e. 73 F dry-bulb, 59 per cent relative humidity, and an effective temperature of 69 deg *without allowance for wind.* The maximum and minimum effective temperatures for the 12 hours are also shown.

In Table B are given average meteorological data for the 5 years 1923 to 1927, both inclusive, computed in the same way.

It is perhaps worth recording that the mean of these 5 years equals approximately the mean of all years for which data are available.

The figures given in Table B indicate that the mean effective temperature for summer outdoor conditions, at Toronto, without allowing for wind, is below the summer comfort zone as proposed in this paper, and that the maximum condition is within the comfort zone.

As it seemed likely that many people, even those concerned with air conditioning, would not realize how different the average outdoor summer conditions in Pittsburgh and Toronto are, the data given in Table C have been collected from the records of the U.S. Weather Bureau and the Dominion Meteorological Office. Other Canadian towns have been included as examples of the different summer conditions experienced.

The data given indicate that, at Toronto, the mean summer temperature is 7 F lower than at Pittsburgh, the mean maximum at Toronto is 8 F lower than at Pittsburgh, and the absolute humidity some 12 to 15 per cent lower. The relative humidity varied, of course, with the temperature, at 8 a.m. it was rather lower in Toronto, at 8 p.m. it was 10 per cent higher.

I suggest, firstly, that one can deduce from these figures that, for the same indoor condition, the reaction of people acclimatized to Toronto would differ from the reaction of people acclimatized to Pittsburgh. Secondly, one can deduce that the application of the comfort chart to summer air conditioning in various cities is very limited; what is considered comfortable by the majority of people depends, to a considerable extent, on the conditions to which they have become acclimatized.

These deductions would point to the need of further study of this subject. Such studies should be carried out in places having average summer outdoor conditions differing from those of Pittsburgh. Two such studies, together with the data presented in Mr. Houghten's paper, might enable us to draw up charts from which comfort zones for the majority of large cities in the United States and Canada could be constructed.

J. W. O'NEILL (WRITTEN): I think that the membership is deeply indebted to Mr. Houghten and the Council for carrying through this work. It is very important to everyone concerned with comfort cooling because the reported results indicate that savings may be effected by changing our conception of required design conditions. If it is true that our present standards for indoor dew point temperatures can be upwardly revised without adversely affecting the comfort effect, it will be possible, in many cases, to reduce costs. If higher dew point temperatures may be used with equal comfort effect, the dew point temperature leaving the conditioner can be raised and this, in many cases, will permit the use of warmer cooling mediums. In the case of mechanical refrigeration systems, the compressor will operate at a higher back pressure resulting in increased output per horsepower. Where water is the cooling medium it may mean that water temperatures from 5 to 10 F higher may be used.

Tests 12 and 25 particularly interested me because of conditions peculiar to Toronto where we have a relatively cool city water supply. For the years 1932 to 1934 inclusive, the average water temperature for June was 45 F, for July 47 F, and for August 52 F. At times it reaches and exceeds 60 F, but this is infrequent, and the periods are of short duration. The duration and frequency of these periods, when the water temperature exceeds 55 F are such that, for the average comfort application, the design may be based on 55 F water. There is, also, usually a compensating feature in Toronto weather which aids the use of city water. We usually do not have prolonged hot spells; but a series of fluctuating hot and cool ones. High outdoor temperatures for 3 or 4 days are usually followed by a few days at 72 to 75 F.

The fluctuations in city water temperature usually lag behind the outdoor changes so that during 85 to 90 F weather, the water is below 55 F. When the water starts to climb towards 55 F the outdoor temperature starts to drop and we have, for 2 or 3 days, a reduced cooling load compensating for the increased temperature of the cooling medium.

For theatre work I have been using as a design base, the following conditions:

Outdoor conditions:	90 F, D.B. 72 F, W.B.
Indoor conditions:	76 F, D.B. 66 F, W.B. 60 per cent R.H. 71.7 deg E.T.
Supply Air Condition:	62 F, D.B. 59 F, W.B. 57 F, D.P.
Metabolic Rates:	Sensible heat 250, latent heat 150, total heat 400 Btu per person.

For an outdoor temperature of 90 F The Guide 1935, in Table 2, page 48, states that the indoor condition should be:

78 F, D.B. 64.5 F, W.B. 57 F, D.P.

and furthermore, it says that the indoor dew point should be maintained constantly at 57 F regardless of outdoor conditions.

Only mechanical refrigeration would fill this bill and some form of reheating would be essential. City water would not be adequate. Test No. 12 was made with conditions maintained as specified in Table 2, page 48 of The Guide 1935, and Fig. 1 of this paper shows that a No. 4 comfort index was attained 60 min after entrance. To provide the specified conditions, 57 F would be required and this would eliminate city water as a refrigerant.

Test No. 25 shows an outdoor condition of:

86 F, D.B. 72 F, W.B.

and an indoor condition of:

76.9 F, D.B. 67 F, W.B. 62 F, D.P. 61 per cent R.H. 72.5 deg E.T.

Fig. No. 1 shows that a No. 4 comfort index was attained in 40 min. I assume, therefore, that these two widely different indoor conditions provided equal and satisfactory comfort feelings insofar as these particular subjects were concerned. If the comfort feeling reaction of the average theatre goer is approximately the same as the reactions of the subjects used in these tests, it means that warmer refrigerants may be used than would be required to maintain the conditions of Test No. 12 and those specified in Table 2 of The Guide.

If the indoor conditions, as used in Test No. 25 will provide comfortable conditions, then 55 F water would be permissible and in the case of direct expansion conditioners, using finned coils, the evaporator temperature could be 50 F instead of the usual 40 or 41 F. In the case of double heat transfer systems, the cooling water could be 55 F instead of the usual 45 F and the evaporator backpressure could be correspondingly raised with a consequent increase in output per horsepower.

The results plotted in Fig. 1 attracted my attention because the conditions specified in Test 25 are almost identical with the conditions that I have been using for theatre work. In Test No. 25 the indoor dry-bulb is practically 77 F and the indoor wet-bulb 67 F, whereas I have been using 76 F, D.B. and 66 F, W.B. so that the two conditions are almost identical. The advantages of the Test 25 conditions as compared to those of Test 12, are more fully appreciated by calculating the respective required supply air conditions for a given case.

Neglecting building load, which is a variable, and can easily be added, and using the load due to people and outside air in a 1000 seat theatre, an interesting contrast is derived from the following design basis.

1. A saturated air condition leaving the conditioner is assumed.
2. Metabolic rates of 250 sensible heat, 150 latent heat, and 400 Btu total heat are used.
3. A saturated air supply of 90 lb of air per hour, per person, is assumed. This amounts to 20 cfm per person.

For Test 12 conditions, the required indoor wet-bulb temperature is 64 F and the total heat is 28.93 Btu per pound of air. The total heat removal per person is 400 Btu per hour. The weight of air is 90 lb per hour per person, so that the required leaving air condition is:

$$28.93 - \left(\frac{400}{90}\right) = 28.93 - 4.45 = 24.48 \text{ Btu per pound total heat in the supply air.}$$

The corresponding wet-bulb temperature is 57.4 F. As the air leaving the conditioner is assumed to be saturated, the dry-bulb temperature, of course, is also 57.4 F. Therefore, for the conditions of Test 12 the required saturated condition leaving the conditioner is 57.4 F.

Test 25 conditions call for an indoor wet-bulb temperature of 67 F having a total heat of 31.15 Btu per pound. The total heat to be removed remains the same as for Test 12 conditions, i.e. 4.45 Btu per pound of air. The total heat per pound of room air is 31.15 Btu and deducting the 4.45 Btu leaves a total heat of 26.7 Btu per pound total heat in the supply air leaving the conditioner. The corresponding saturated supply air condition is 60.8 F. The temperature level for Test 25 conditions is 3.4 F higher than for Test 12 conditions.

The required leaving air temperatures are 57.4 F for Test 12 conditions and 60.8 F for Test 25 conditions, a difference of 3.4 F. This may not seem much until the total load requirements are calculated.

Suppose that an outdoor air supply amounting to 10 cfm per person is used. This amounts to 50 per cent of the total air supply and if the outdoor wet-bulb temperature is 72 F the total heat to be removed per pound of outdoor air is:

For Test 12 conditions $35.17 - 24.48 = 10.69$ Btu per pound
For Test 25 conditions $35.17 - 26.7 = 8.47$ Btu per pound

The fresh air and people loads for 1000 people are:

Test 12 conditions 681,000 Btu per hour = 56.8 Tons
Test 25 conditions 581,000 Btu per hour = 48.5 Tons

The difference is 100,000 Btu per hour or 8.3 Tons

But this is not the ultimate saving, because if city water at 55 F is used the water consumption is dependent upon the allowable temperature rise which, in turn, is dependent upon the leaving dew point temperature.

Taking the case of a single stage parallel flow washer, the leaving water could not leave at a temperature higher than one degree lower than the required leaving dew point temperature. For Test 12 the leaving water temperature might be 56 F approximately; a rise of only one degree. The water consumption would be $681{,}000 \div 600 \times 1 = 1135$ Imperial gallons per minute. This water consumption, of course, would be entirely impractical.

For Test 25 conditions, the water might leave at approximately 59 F; a rise of 4 F. The water required would be $581{,}000 \div 600 \times 4 = 242$ Imperial gallons per minute. Even this water consumption is high but is reduced approximately 79 per cent from that required for Test 12 conditions.

In the case of coils, the leaving air condition would be approximately 90 per cent saturated, so that the leaving condition would be 62 F, D.B., 60 F, W.B., 59 F, D.P. for Test 25 conditions. The terminal temperature difference at the leaving air side would be 7 F. A mean temperature difference of approximately 11 F results in a reasonable amount of surface and this allows a water temperature rise of 12 F. The water would enter at 55 F and leave at 67 F.

The water consumption for Test 25 conditions would then be based upon a 12 F rise in water temperature and would amount to 81 Imperial gallons per minute, which is entirely practical and very economical. The water sells for approximately

11.7 cents per thousand Imperial gallons so that the operating cost would be 57 cents per hour.

In a case of direct expansion coils Test 12 conditions would most likely be obtained with 41 F evaporator temperature, whereas Test 25 conditions could be obtained with 50 F evaporator temperature, without using an impractical amount of surface. The resultant increase in compressor capacity is of the order of 18 per cent. This increase in compressor capacity would, in many cases, allow the use of a smaller condensing unit. For instance, if dichlorodifluoromethane at 41 F suction temperature were used for the 681,000 Btu per hour load, as of Test 12 conditions, four 15-hp machines rated at 167,300 Btu per hour with 60 F condensing water, would just be a little too small, having a total capacity of 670,000 Btu per hour. If Test 25 conditions were used, the load is 581,000 Btu per hour. Three 15-hp units at 50 F suction temperature, and 60 F condenser water have a capacity of 592,000 Btu per hour.

In presenting the foregoing figures, I hope that the impression is not created that I construe Mr. Houghten's work to mean that the relative humidity can run wild. There is a very real danger that these data will be so construed and the result will be a chaotic competitive condition, with bewildered purchasers. This condition is bad enough now. Purchasers are offered, under the magic name air conditioning, everything from a couple of spray nozzles in a tank to a complete refrigeration plant, and the cheaper the system, the more elaborate become the claims and guarantees. In the foregoing figures, therefore, I would like to repeat that the worst condition I have used provides a 72.5 deg E.T. as of Test 25.

The highest effective temperature that I could find plotted against comfort index was 73 deg. I would like to ask Mr. Houghten if from his experience, he thinks 73 deg E.T. should be the upper limit for the conditions which prevail in the average movie theatre where the occupants are of both sexes, all ages and all physical conditions, in contrast to the subjects used in this work who apparently were all young men in good physical condition.

H. H. Angus (Written): Fig. 8 shows a fluctuating condition. Is there any explanation of this condition?

In Table 2, Tests 39 and 14, a difference of 30 min is shown in column *Q* for a comparatively small change in outside conditions and uniform inside conditions. Similar results are shown between tests 36 and 8 but between tests 67 and 74 there is little difference for a wide change in outside conditions. From these values it would seem that item *Q* varied considerably independent of indoor and outdoor conditions.

R. B. Crawford (Written): Past experience has pointed out that research in the field does not verify laboratory studies on many phases of air conditioning. The precision of measurement and the law of averages is seldom attained on a laboratory scale which would be comparable to a full sized installation. Also artificial outdoor weather conditions do not serve to acclimatize a person to the proper physiological and psychological reactions which extended periods of exposure to normal summer weather would do.

It is my experience that the climate, class and habits of the people of any city will vary the conditions they desire and no standards can be adopted which will suit many towns. Phoenix, Ariz., for example, has a number of installations we engineers would call ideal but which are considered gross failures by the native Arizonian acclimatized to low humidities and high temperatures. I doubt from observations between here and Baltimore whether the same conditions in both towns would promote air conditioning development.

Conditions desired for comfort depend entirely on the type of job. In a theatre the radiant heat from massed people demands a much lower temperature for com-

fort than would a residence, hotel or other sparsely populated enclosure. In a restaurant, for example, it is a known fact that lower humidities than are customarily used in air conditioning comfort work retard the transmission of food odors.

It is my opinion that the Society is definitely committed to the effective temperature standard and intends to force its acceptance whether people like it or not. After all, comfort cooling practiced by everyone has been a makeshift compared to our industrial air conditioning jobs and it seems foolish to me to try and pretend comfort is being enhanced if the system is designed to follow some effective temperature standard. After all there are plenty of jobs in Washington which operate above, below and right in the comfort zone, but I fail to see any marked superiority in the customer appeal or the net financial return from the job regardless of what effective temperature they attain. I do not feel that the effective temperature standard is the wrong idea to be used to promote air conditioning development because

1. The layman (and some engineers) do not understand it.
2. The effective temperature is effected by air motjon of various degrees and no one knows what degree he may get until he has laid out and installed the job.
3. The promulgation of the effective temperature idea is hurting business by beclouding the issue in a willing purchaser's mind.
4. If no standards were adopted, every good firm, every good engineer would succeed by virtue of his past performances. Those who tried to skin their jobs by using less outdoor air and more air motion to reach the standard acceptable effective temperature would be doomed to failure.

To get down to the data in the paper, a group of engineers visited the Crown Cork and Seal Plant in Baltimore on July 2-3, 1927, during a protracted heat wave. The conditions were approximately 90 F dry-bulb with 20 per cent relative humidity. Everyone, including the employees, was comfortable and there was no shock either entering or leaving the treated area. According to the Society data they should be very warm and perspiring which was not the case. I would say very definitely that the Society data on low humidity comfort sensations is valueless.

At the other end of the chart we are asked to believe that humidities of 60 F dry-bulb and 85 per cent can be made comfortable if the effective temperature is right. To combat this with facts I offer the many early air washer installations without refrigeration which produced such a condition which were considered failures. Of late years Sears-Roebuck installed a number of jobs supplementing washed air with intensive air motion to reach acceptable effective temperature standards. These jobs, I understand, also failed to produce comfort.

In Persia an air conditioning job based on normal standards was installed in a hospital where those overcome by the heat and humidity were taken for relief. Almost invariably it is reported the victims contracted pulmonary troubles and in many cases were ruined for life as a result of the standards not fitting the job.

E. V. FINERAN (WRITTEN): From the test data given in the paper, Fig. A was prepared, which seems to indicate that there is a relation between the intensity of the shock experienced and the difference between the effective temperatures of the conditioned space and outdoors. In making the graph, Column M, Degree of Cold Shock Upon Entry—Test Room, was subtracted from Column L, Degree Feeling of Warmth Before Entry—Test Room. The resultant figures were assumed to be proportional to the intensity of the shock. The majority of the points indicated that there is a straight line relation, the greater the effective temperature difference the greater the shock. It is interesting that 4 of the 5 points which did not line up with the rest were for tests in which the inside relative humidity was equal to or greater than the outside relative humidity.

In this discussion, it was attempted to show that for equal effective temperature differences the intensity of the shock was proportional to the inside relative humidity. There were few data given upon which such a study could be based; however, in the two cases in which data were given for the same effective temperature differences and with high and low relative humidity the shock was smaller for the lower relative humidity.

Pursuing further the method employed in the paper and referring to curve B it can be reasoned that the shock upon entering a conditioned space is related to the dry-bulb temperature difference between the outside and inside conditions assuming constant relative humidity in the same way that the change in latent heat given off by the body is also related to the difference in dry-bulb temperatures.

Assuming that there is a relation between the intensity of the shock and effective temperature difference as indicated in Fig. A then, if we are dealing with a constant

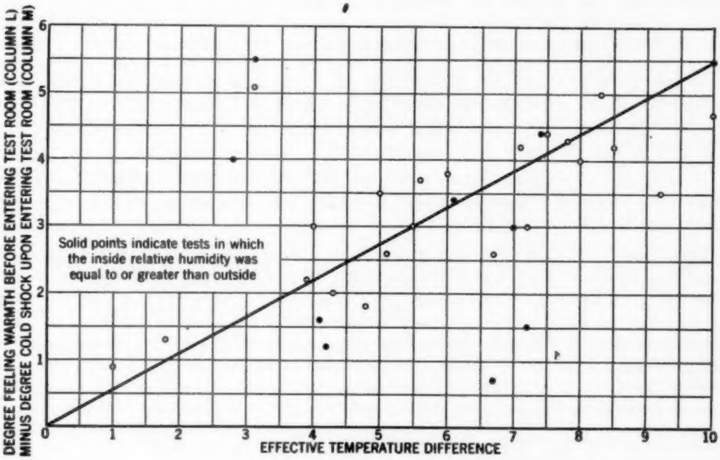


FIG. A. RELATION BETWEEN INTENSITY OF SHOCK EXPERIENCE AND EFFECTIVE TEMPERATURE DIFFERENCES

relative humidity condition both inside and outside of 50 per cent, for each different dry-bulb temperature we will have a different effective temperature and a different degree of shock upon entering from a constant outside condition.

Looking at it another way, no one would hesitate to say that there would be practically no shock for a 1 or 2 deg dry-bulb temperature difference at constant relative humidity and that as we consider slightly larger and larger temperature differences the intensity of the shock should increase gradually.

The intensity of a shock upon going outdoors from a given conditioned space is probably related to the intensity of the shock upon entering the conditioned space from outdoors.

We have then this relation between intensity of shock and dry-bulb temperature difference and this other relation between the change in latent heat emission of the human body and change in dry-bulb temperature difference. In the absence of proof to the contrary may we not assume that degree of shock is related to the magnitude

of the adjustment necessary in our heat control mechanisms to come to equilibrium with the new conditions.

Both shocks occur because of the human body's inability to instantaneously adjust its metabolism to any variation in atmospheric conditions. Therefore, if we can avoid the necessity of making a radical adjustment necessary we can reduce shock.

As the proportions in which sensible and latent heat are given off by a body are a function of the dry-bulb temperature it is possible within limits to maintain the desired effective temperature in the conditioned space and at the same time necessitate only a minimum readjustment of our heat control mechanisms upon entering or leaving.

The means of accomplishing this highly desirable condition is low relative humidity. For a given effective temperature the lower the relative humidity (high dry-bulb temperature) the greater the proportion of latent to sensible heat given off by a body. Therefore, for a given effective temperature the lower the relative humidity the less the shock upon either entering or leaving the conditioned space.

To more fully reveal the possibilities of such, a procedure for increasing the acceptance of air conditioning especially in stores and other establishments where the patrons enter for short intervals only is worthy of thorough and immediate investigation.

It is suggested that the future study of this question contain more data on tests with inside effective temperatures above 72.5 deg and with greater differences between inside and outside relative humidities.

In a study of this nature the data obtained cannot be expected to have mathematical accuracy. Therefore, it is advisable to have a check upon the accuracy of each test. In order to avoid unnecessary duplication it is suggested that the results of each test be plotted upon as many different graphs as can be found to indicate a relation between the results, so that points which are out of line on many of these different graphs may be rechecked. In the data contained in the paper such a procedure indicated several tests which would be worth rechecking. The suggestion that skin temperatures be studied in this connection is a good one.

It is stated in the paper that "relative humidities in the neighborhood of 20 per cent and around 75 deg effective temperature occasionally gave a dry burning sensation immediately after being entered, particularly when entered from a cooler condition." In this connection it should be emphasized that the usual procedure would be to enter the low humidity condition from a warmer and more humid atmosphere in which the occupants have been perspiring freely as in the data given in the paper for the conditions prior to entering the test room. Also 20 per cent is probably a lower relative humidity than would normally be used in the field. Therefore, we can say that we may never expect discomfort from low relative humidity.

R. C. JORDAN (WRITTEN): The results reported in this paper are a stride towards the elimination of complaints by the public caused by their uncomfortable reactions to summer air conditioning. The evidence presented that the effective temperature of the indoor condition is of much greater importance than the moisture content of the air inside, or the outside dry-bulb temperature, should result in a change for the design basis of many future installations. However, the authors' conclusion that, "the study indicates rather conclusively that the average person will become comfortable within 20 to 40 min after entering any atmospheric condition within an effective temperature range of 70 or 71 deg to 74 or 75 deg," appears to be dangerously inclusive for the reasons which follow.

The subjects were all male, between the ages of 19 and 23 years and in normal health as regards "body temperature, pulse rate, respiration and other physiological reactions which might affect their temperature regulation under various atmospheric conditions." The authors mention in one place that older people with less vitality might react differently, yet they do not warrant this important enough to modify their conclusions. Obviously such subjects are not average, and even if they were, comfort standards such as these should give consideration to individuals who are not of average health and who might be subjected to shocks possibly dangerous to their health. It is doubtful whether the majority of the theatre box office complaints are from men between the ages of 19 and 23 and in normal health.

The clothing worn during the tests by the men included a light weight summer coat. Not all men wear summer coats when the dry-bulb temperature is at 95 F. Those who do not will of course be subjected to a greater shock upon entering a cooled enclosure. The ladies in the audience will certainly not be as heavily clothed as the subjects were in these tests.

Only 5 subjects were used for the determination of the data from which all the conclusions were drawn, and these subjects under no conditions could be considered a good cross-section of the theatre-going public. The authors were dealing with data which required a large number of cases before the means they determined could be considered representative of the frequency distributions with which they were dealing. No probable errors were calculated, and it is therefore difficult to tell what the reliability of those means were.

The authors inferred that as long as the effective temperature is between 70 to 75 deg, the relative humidity can be allowed to vary as high as 85 per cent. While this may be true as far as the feeling of warmth is concerned, there is some doubt as to whether such high humidities should be included in comfort standards without further research. There is some evidence of physiological reactions to high and low relative humidities other than those of warmth.

The feeling of warmth of the subjects has been determined and recorded in accordance with an arbitrary numerical scale as follows: (1) cold, (2) too cool for comfort, (3) comfortably cool, (4) ideally comfortable as far as the feeling of warmth is concerned, etc. There is no evidence that the increment of warmth between (1) and (2) is the same as that between (2) and (3), yet the authors have graphed these data on the basis that all the increments throughout the scale are equal. This tends to distort a number of the curves presented and thereby misrepresent the data.

J. R. Roberts (Written): It is my feeling that these tests were not conducted upon subjects who were sufficiently representative of the mass of people who are being subjected to air conditioning. Also these subjects were too conscious of the fact that they were being used as guinea pigs and the results show it. (Fig. 1, all tests end with the subjects feeling ideally comfortable to the extent that the results show one straight line.) In the average air conditioned office, two people at adjoining desks will feel too warm and too cool respectively while being subjected to exactly the same conditions.

In test 44, the subjects never reached a feeling of comfort; yet in test 40, the subjects were comfortable within 20 min. This is difficult to explain because the subjects in test 40 were "too warm for comfort" before entering and had a perspiration rate of 1.4 while the subjects in test 44 were only "comfortably warm" before entering and had a perspiration rate of 1. The outdoor dry-bulb was the same in both cases and the indoor effective temperature varied by only 0.8 of a degree.

In test 27, the subjects entered from an environment where they had felt "too hot." The main feeling before comfort was reached was one of coolness. The degree of perspiration was 2.2 and yet they were comfortable within 20 min. In test 12, the

subjects entered from a point of practically the same dry-bulb into a space of the same effective temperature. The rate of perspiration was the same and yet instead of feeling cool before comfort was reached, they felt warm. The entering contrast was less and yet where in the previous test the subjects were comfortable in 20 min, this time they required 35 min.

Comparing tests 12 and 37, the subjects enter from identical dry-bulb temperatures with the same rate of perspiration. For some reason not explained, the subjects in test 37 feel more uncomfortable before entering; yet they reach a feeling of comfort in 30 min while the subjects in test 12 require 55 min to arrive at the same condition. Also the subjects in test 12 never experience a feeling of cold shock. Surely we are not supposed to deduce from this that the greater the contrast at entering, the less time required for the body to adjust itself. This does not hold true in several of the other tests.

A set of comfort standards for summer air conditioning is very desirable. However, it would be a mistake to publish a new set of standards which was not representative of the reactions of the general public who will live in the air conditioned spaces. Since the subjects of these tests were not representative people and since the tests contradict themselves a great deal, I believe that it would be far better not to present this information to the general public until more conclusive tests are made. The general public at the present time is very apprehensive about air conditioning. Changing the standards now, and then revising them again next year, after more complete tests are made, would only be confusing.

H. M. BETTS (WRITTEN): A careful study of results obtained in this test must lead one to the conclusion that the relative humidity existing within an enclosure provided with summer cooling is relatively unimportant from the standpoint of comfort of the occupants provided the effective temperature is maintained within very definite limits.

It must be observed, however, that these investigations are not complete. Continued study along these same lines, particularly when applied to conditions approximating those existing in theatres and large auditoriums may well result in the establishment of more definite relative humidity limits.

It is to be hoped that the authors of this paper will be given the opportunity to make more extensive studies of this subject for the purpose of establishing definite limits, thereby eliminating much of the dissension and resulting confusion which apparently now exists. If these data are to be of value, they should be based on observations of subjects of both sexes and various ages and should be obtained under conditions and in surroundings comparable with those met with in actual practice.

F. B. ROWLEY: This paper is a timely presentation upon a subject about which there is much controversy. There are naturally several points for discussion, many of them no doubt requiring further experimental work for an adequate solution.

In a subject so elusive as proper air conditions for occupied spaces it is dangerous to draw hasty conclusions. As the author states, more research work must be done and certainly the results thus far obtained should be checked with a wider range of test subjects before putting any of the findings into practice or allowing them to be used as standards.

From Table 2 the extreme outside dry-bulb temperature was 95 F with a relative humidity of 52 per cent, and in a great majority of the tests the air conditions were much more moderate. In a final analysis it would also be desirable to consider other geographic locations.

From test data shown in Fig. 2 there are several observations which might be made. For those tests which were made by varying the effective temperature first

in one direction and then in the other there is usually a different boundary line established for the maximum or minimum comfort conditions. From the test results it is difficult to draw lines which will give the upper and lower limits of comfortable effective temperature as obtained. If, however, a line is drawn from midway between 76 and 77 on the saturation line to a point corresponding to 88.5 on the dry-bulb temperature line, this gives a reasonable average for the upper limits established. Likewise if a line is drawn from 71 F on the saturation line to 77 F on the dry-bulb temperature line it gives a reasonable lower limit. These lines have a much deeper slope than the effective temperature line shown on the chart and indicate the need for more test data.

In tests Nos. 63 and 66 of Fig. 5, the test subjects were taken from air of identical conditions and transferred to air of the same effective temperatures, but with a low dry-bulb and high relative humidity for No. 63 and high dry-bulb and low relative humidity for No. 66. The curves show that at the end of an hour the test subjects felt substantially the same comfort in either condition, but that it took a much longer period to reach a comfortable condition in the high relative humidity air of test No. 63 than it did in the low relative humidity air of test No. 66. This was undoubtedly due to the slow evaporation in test No. 63 as compared with test No. 66, and indicates that at least for the first hour the occupants did not feel the same degree of comfort even though the effective temperature was 73 deg in both cases.

By comparing tests Nos. 67 and 74 as shown in Fig. 4 it will be noted that the test subjects remain warmer in test No. 74 than in test No. 67 throughout the period of 170 min, even though the effective temperature is 75 deg in both cases. The main differences in the condition of these two tests is that the subjects were preconditioned in an atmosphere of 9 F higher dry-bulb temperature for test No. 74 than for test No. 67. The results of these tests would indicate that the effective temperature of conditioned air should be maintained lower if the occupants are to come from high outside air temperatures. This is contrary to conditions found in general practice.

The question of relative humidity is of such importance that it would seem inadvisable to draw definite conclusions until more data are available.

L. T. Avery: I should like to inquire of Mr. Houghten the quantity of air circulated in the test room, the over-all air change, and if measured, the actual air velocity. Also, were the subjects far enough apart so there was no radiant heating effect between them, and was the lighting shielded so there would be no radiant heat from the lights? We are indebted to the investigators who have prepared this paper for clearing up not so much chaos as criticism of our industry by those who say we have been introducing air too cold. We have found in actual experience in all of our work that relative humidities of 60 per cent are comfortable. It has been the criticism of the lower temperatures that has caused the doctors to look doubtfully at our efforts in air conditioning for health.

Referring to the charts, I note that the higher temperature levels were taken at the end of August. We have found that in starting a system in the spring we must set the controls for a lower effective temperature than in the fall. In a climate like Chicago, the human body adjusts itself during the summer season and will tolerate higher temperatures in the late summer than it will in the spring.

It is my opinion that the high relative humidity that comes with the warm summer weather sets up an internal physiological reaction so we can stand higher humidities without perspiration late in the summer, as contrasted to earlier in the spring.

It would seem desirable to perform tests earlier in the season with the higher humidities to observe if there is not a relationship between our adjustment and the outside weather.

I believe we have determined that there are probably not lines of equal comfort but there is a comfort zone that is fairly wide in range.

It is my opinion that we will find that as the relative humidity is increased or decreased, say if it is increased above 60 per cent or decreased below 40 per cent, we will find the individual variations greater, and the individual physiological response ceases to be subject to averaging. I would urge that we take into consultation on further tests of this kind physiologists and physicians who are capable of correlating the data to assure us that we are properly conditioning for health. This factor seems to me the weakest part of the picture. We are setting up comfort standards and talking about health, but as yet we have not proved anything about health.

MR. HAAS: Mr. Houghten made mention of the fact that the question of varying conditions, which are on the effective temperature line, such as high humidities, low temperatures, low humidities and high temperatures, while they were equally comfortable might not be equally healthy. We do not have a lot of data but I would be interested in hearing Mr. Houghten's comment on this subject.

J. N. HADJISKY: I notice that the speaker mentioned that the subject had to take a half-hour walk before the test.

Another thing of importance that seemed to have been left out, or not mentioned, is the fact that in summer time the kind of food that one eats has a considerable effect upon the metabolism for the succeeding 2 hours, and especially with certain foods rich in hydrocarbons. To what degree was this item considered in the subjects under test?

J. J. AEBERLY: I had occasion to discuss this paper in a meeting of the Illinois Chapter. At that time it occurred to me that, when data of this kind are presented there is much confusion and a tendency to draw conclusions based on our limited experience rather than on the data presented. This study will not settle the moot question in the Society relating to the standard for comfortable conditions in the summer time, when air conditioning is used.

If you will refer to Table 1 and Fig. 2, you will note that the Society has set up standards for summer cooling based on the dew point temperature and these values are not consistent with the data presented in the paper and plotted in Fig. 2. Is the dew point the most satisfactory standard for expressing the range of reasonable comfort in an auditorium, or is some other range of temperature? One of the previous speakers, I think, presented this very nicely. He said: "We are agreed on a zone of satisfactory conditions." What is the best measure in terms which the Society knows to express this range?

If you will refer to Fig. 2, you will see that the points represented by Line A-A are those contained in the tabulation published in THE GUIDE and recommended to the industry. It indicates constant dew point. This work shows that we must proceed on laboratory findings and not on practical experience. The data in this paper show that effective temperatures run most consistently parallel with equal expressions of warmth as established by the normal person. We should at this time, in the light of these facts, decide to correct THE GUIDE, eliminating the tabulations recommended and substituting effective temperature range.

MR. HAAS: If we lose the heat from our skin through perspiration is that as healthy as if we lose it all by the skin? I would like to get a comment from some one who is dealing a great deal with this subject.

F. C. HOUGHTEN: Professor Rowley's question really involves the accuracy of the direction of the effective temperature lines. It is true that with low relative humidities the subjects were apparently comfortable at a little lower dry-bulb

temperature. If that is universally true, then the direction of the effective temperature lines as they were determined at the laboratory about 12 years ago are slightly in error.

I would not in any sense attempt to change the direction of those lines based upon this study, for the original study made 12 years ago to fix the direction of the lines was more consistent, thorough, and better designed to establish that fact. After all, there were not many tests in this study which could be used to re-establish the direction of these lines. If we accept the maximum indicated change in their direction from this study, it would not change the direction of a line more than one degree for a relative humidity range from 30 to 100 per cent.

Mr. Avery asked concerning the quantity of air change and the air movement within the room. There were always supplied at least 20 cfm of outside air per occupant.

The air velocity within the room was somewhat greater than that used in originally establishing the effective temperature lines, when velocities ranging from 15 to 25 fpm were used. In this study the velocities ranged from 25 to 40 fpm, as measured by the Kata thermometer. The reason for the higher velocity in this study was to maintain uniform conditions throughout the room with cooling instead of heating. It should be remembered, however, that this entire velocity range is within the confines of the specifications of the Society's Ventilation Standards Code.

Mr. Avery also asked regarding the distance between subjects. They were about 4 ft apart, which allowed less radiation effect from one person to another than would pertain in this audience, or in a theatre or other audience hall.

He also asked concerning the radiation from lights. In some of the early tests there was a 100 watt electric light bulb about 3 ft from the head of the nearest subject who distinctly felt its warming effect. In all subsequent tests the smaller lights, scattered throughout the room and farther from the subject, were used.

Mr. Haas asked concerning health effect resulting from different conditions. That subject has never been investigated, at least, not by the Laboratory, and I do not believe by any one else. It is always hazardous for any one engaged in research to offer opinions. However, for what it may be worth, my own opinion, based not on data collected for that purpose, but on general observations made during my studies at the Laboratory, is that a person's health is not adversely affected by high relative humidities. I would just as soon exist in an atmosphere of 70 or 85 per cent relative humidity, provided the temperature is correct and things are kept clean and sanitary. Remember, however, that such high relative humidities may result in things mildewing and other unsanitary conditions, unless precautions are taken to avoid them.

Low relative humidities, from my own observations, are undesirable, particularly in winter heating, because of greater susceptibility to colds. This subject should be answered by research. Such a study must be unusually comprehensive and must be made over a long period of time, with many subjects, so as to rule out variables.

Some one asked specifically concerning losing heat from the human body, as evaporation or as sensible heat loss. Within reasonable limits, I do not believe it makes any difference, so long as a person is comfortable, that is, neither too hot nor too cold. A person is not comfortable if he is wet with perspiration. To get a great loss in latent heat except with very low humidities a person must be wet with perspiration.

Mr. Aeberly rightly pointed out that the Laboratory findings should not be taken directly in deciding upon practical applications. The Laboratory should be a fact-finding organization, working with committees in the field who have knowledge of

practical applications in order to arrive at workable solutions which are practical and, at the same time, based on fact.

Mr. Aeberly further discussed the question of dry-bulb versus wet-bulb, versus effective temperature. That is an old question which is still discussed. The best you can say is that over a certain region plotted on the psychrometric chart people are comfortable. Across that region you can plot any kind of a curve of comfort. As has been shown, a short section of the 57 F dew point line falls within that region. A certain length of some certain dry-bulb temperature line can be plotted across that zone, but it will run out of the zone either at the top or bottom, or, if properly selected, at both ends. Likewise, a wet-bulb temperature line can be chosen which traverses the zone of comfort, but any wet-bulb line chosen will run out at the top and bottom; that is, if you take a given wet-bulb line, its lower dry-bulb temperature limits will be too cold, while its higher dry-bulb limits will be too warm.

Any argument raised concerning these three factors is more or less an academic question. It does remain, however, as established by facts found in many, many studies, including this one, that effective temperature is by far the best index of the feeling of warmth.

W. L. FLEISHER: I am sorry I was not here at the beginning of this discussion because, in a way, this is my special interest, and I also think it should be of great interest to the Society, because, as you realize, whatever we decide or discover will greatly affect the spread of air conditioning.

I do not know whether Mr. Houghten in presenting this paper brought out the point that the occasion for this Committee arose because of the fact that the apparatus requirements for maintaining a 57 F dew point required under certain conditions for very little additional comfort, greatly increased air conditioning apparatus.

Neither do I know whether Mr. Houghten said that during the past year as chairman of this Committee I had the opportunity of investigating much higher relative humidities than the ordinary comfort zone indicated as comfortable with great success. I think that the whole discussion and the interest in this particular phase of the subject requires additional investigation. I know that Mr. Houghten has spoken before a number of the Chapters on this subject during the past year. I myself spoke on this subject in Washington, Boston and New York, and the great interest that lies in the work which is being done shows that the continuation of this work must be carried on to a much greater extent.

I had a very interesting experience not over 2 or 3 weeks ago on investigation of comfort zone conditions, entirely outside of any possible indoor results. I have just returned from the West Indies. On the boat, I made hourly tests after we got into the tropics, of conditions which are entirely out of the comfort zone, but with an air movement which is entirely out of our possibility of reproduction. With conditions as high as 78 F dry-bulb and 85 per cent relative humidity with a very strong breeze nobody felt uncomfortable, and I questioned everyone on the boat.

This Committee has been continued and the work which it is carrying on we want to broaden in the field, as well as the laboratory, as much as we can.

Now as to the loss in heat of the human body, which is one of the questions that was brought up at this meeting. I find among the medical profession with whom I have discussed this matter, that there seems to be uniformity of opinion that it does not make any difference how you lose the heat from the body, by evaporation or by radiation and convection, so long as you can get rid of it.

Dr. Mills, in the discussion of this point said, "After all, in the winter you lose practically nothing by evaporation and everything by convection, and radiation," and as far as he was concerned he did not think it made any difference how the heat was

liberated as long as it was lost. If you lost the same amount of heat in the winter by evaporation as in the summer, you would be in a terrible state, due to drafts and cold and other things not conducive to comfort.

The idea in presenting this subject was to broaden the scope of air conditioning. Furthermore it seems that the whole subject of health and comfort should be co-ordinated, probably under this Committee or an enlarged committee, for it is doubtless one of the most important projects the Society has confronting it at the present time. I am anxious that fixed codes for air conditioning, such as have been prepared by the *Chicago Association of Air Conditioning Engineers,* and as are suggested for other cities and have been suggested for the Society as a whole, should not be too conclusively arrived at or too strongly urged on the public, before we are absolutely certain that those conditions which have been at least indicated by the work this year are thoroughly tested.

Anything that we can get from the members of the Society or allied groups who have opportunity of investigating conditions of comfort outside our Committee are so important I do not think we should arrive at any conclusions or be biased either by the results already determined or by some results that may be arrived at which are outside our present understanding.

I urge on the Society that the work of this Committee should be given the co-operation of every member, because, there is enough in the work that this group is undertaking on which to found a whole Society. I feel that the comfort of the people, which is the great cry of today, and which has received so much accord from the people at large, is the subject that appeals to the public.

Mr. Aeberly: I wonder if Mr. Fleisher would be kind enough to answer one question. Does your committee contemplate taking any action due to the result of this study with reference to The Guide for 1937? Does it intend to remain silent on this subject, or will it make some definite recommendation as to what method should be substituted?

This paper shows that we are giving a false impression to the trade, namely, that there is but one measure, when in fact there are a number of them, if values are established by the processes used in obtaining the tabulation published in The Guide.

Mr. Fleisher: I do not know whether the Guide Publication Committee will be guided by unfinished investigations or not.

We have another problem in the Society which has been unanswered for years and that is only second in importance to this subject. It is second in importance because in a way we have so much background that we are not definitely upset or affected by it. This problem is the effect of loss of heat from buildings in the wintertime by wind velocity.

For years, in The Guide, we had a factor of safety for wind velocity, as against an impeccable rule that the heating contractors had advanced, which allowed one degree drop in temperature for each mile of wind. That was never accepted by the Society, nor included in The Guide, but during the time I was chairman of The Guide, we arbitrarily dropped the exact factor of safety, and there is now no mention of it, except reference to using one's judgment. I think it is a mistake to leave such matters to the judgment of the public.

Because this is a tremendous subject, and until we have more information than we have at the present time, I do not believe that the effective temperature lines, and the comfort zone, should necessarily be changed in The Guide.

The conditions which have been referred to have done very little harm. The enlarging of the comfort zone for different conditions than are indicated in The Guide would very definitely affect, for one thing, the refrigeration requirements.

If enough data are obtained this year I would suggest that in those chapters affecting the comfort conditions, at least mention of the work should be made. I think undoubtedly some of the work that has been done should definitely be indicated in THE GUIDE, and I would make that recommendation.

We do not know what the Guide Publication Committee is going to indicate for this next year, but certainly from the standpoint of our present findings, and from a great deal of field work that has been done, I would definitely recommend that some of the conditions which are being called for in the various standards of the various cities should certainly be at least augmented.

To indicate that all of the work which has been done has not been entirely laboratory work, I would like to mention the fact that I had the opportunity of studying conditions this last year which were on the effective temperature lines, but as high as 80 and 85 per cent relative humidity in various enclosures in which probably two million people were at least generally questioned as to their feeling of comfort. The number of people who complained of the higher conditions was almost negligible compared with those people who distinctly felt comfort and less shock than ordinarily was felt in low relative humidity conditions. This type of work would indicate very definitely that we really should not proceed with the dogmatic statement of what are the right conditions until these investigations are carried still further.

No. 1036

SEMI-ANNUAL MEETING, 1936

THREE hundred and fifty members and guests accepted the invitation of Philadelphia Chapter to enjoy the technical sessions and the entertainment program provided for the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, June 22-24, 1936, at The Inn, Buck Hill Falls, Pa.

One of the enjoyable features of the meeting was the joint session held with members of the *American Society of Refrigerating Engineers,* who were meeting at the same time with headquarters at Skytop Lodge nearby.

Two meetings of the Council were held on Sunday, June 21, and on Wednesday, June 24. A luncheon and meeting of the Committee on Research occurred on June 22, a breakfast meeting of the Nominating Committee took place on June 23, and the Guide Publication Committee had a luncheon and meeting. Other technical and general committees held conferences during the three-day session. Amendments to the By-Laws were adopted, nominees were selected to serve on the Committee on Research and the Nominating Committee selected its slate for 1937.

The Semi-Annual Meeting 1936 was called to order by Pres. G. L. Larson, Madison, Wis., Monday morning, June 22, at The Inn, Buck Hill Falls, Pa., and Wilbur Smith, president of Philadelphia Chapter, welcomed members, guests, and ladies to the Poconos and expressed the pleasure of Philadelphia members for the opportunity of serving as hosts. A brief response was made by 1st Vice-Pres. D. S. Boyden, Boston, Mass.

Amendments to Constitution and By-Laws

The first order of business was consideration of the following amendments to the Constitution and By-Laws, which were presented by R. H. Carpenter, New York, N. Y., chairman of the committee.

Article C-II, *Section 2.* The membership of the Society shall consist of Honorary Members, Members, Junior Members, Associate Members and Student Members.

To Be Amended as Follows: The membership of the Society shall consist of Honorary Members, Life Members, Members, Associate Members, Junior Members and Student Members.

Article B-II, *Section 5.* Any person who has been a member of the Society for fifteen (15) years or more and has retired from business, then upon reaching the age of seventy (70) shall have his dues remitted for the current year and for ensuing years, without surrendering any of the privileges of membership as long as he lives.

To Be Amended as Follows: Any person who has been a member of the Society for 15 years or more, upon reaching the age of 70, shall have his dues remitted for the current year and for ensuing years without surrendering any of the privileges of membership as long as he lives.

Article B-IV, *Section 7:* The dues of a new member of any grade may be pro-rated monthly for the balance of the year but if the amount thus paid is less than five dollars ($5.00) such member shall not be entitled to receive the volume of the TRANSACTIONS for the year in which he is elected, but he shall otherwise be entitled to all the rights and privileges of membership. Forty per cent (40%) of the pro-rated dues of Members and Associate Members shall be considered as a contribution to the Research Fund and shall be immediately deposited in such fund and shall not be used for any other purpose.

To Be Amended as Follows: The dues of a new member of any grade may be pro-rated monthly for the balance of the year but if the amount thus paid is less than five dollars ($5.00) such member shall not be entitled to receive the volume of the TRANSACTIONS or THE GUIDE for the year in which he is elected, but he shall otherwise be entitled to all the rights and privileges of membership. Forty per cent (40%) of the pro-rated dues of Members and Associate Members shall be considered as a contribution to the Research Fund and shall be immediately deposited in such fund and shall not be used for any other purpose.

Article B-VIII, *Section 1.* At the first meeting of the Council after the Annual Meeting, the President shall appoint from the members of the Council the following committees consisting of three (3) members each, to act under the direction of the Council:

(*a*) Executive Committee.
(*b*) Finance Committee.
(*c*) Membership Committee.
(*d*) Meetings Committee.

To Be Amended as Follows: (*e*) Standards Committee.

Add Article B-VIII, Section 6. The Standards Committee shall consider all questions pertaining to the establishment and revision of Society standards and shall report its findings to the Council.

In accordance with Article C-XVI of the Constitution it was voted that the amendment to Article C-II, *Section* 2 of the Constitution be submitted to the membership prior to the next Annual Meeting in the required manner.

Mr. Carpenter moved the adoption of Article B-II, *Section* 5, Article B-IV, *Section 7,* and Article B-VIII, *Sections 1* and *6.* The motion was seconded by J. D. Cassell, Philadelphia, Pa., and when President Larson put the question, the vote was unanimously in favor of adoption.

On Tuesday morning members of the *American Society of Refrigerating Engineers* and the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS gathered for a joint session in the auditorium of The Inn. The officers of both societies were on the rostrum when President Larson called the meeting to order.

W. H. Carrier, Newark, N. J., past president of both societies, was introduced and acted as chairman. The *American Society of Refrigerating Engineers* contributed two technical papers, one of which was presented by Prof. C. W. Chamberlain, East Lansing, Mich., entitled, Theory and Practice of the Heat Pump. In the absence of D. C. Morrow, Washington, Pa., his paper on Water Supply as Affected by the Demand for Summer Air Conditioning was presented by title.

On Tuesday, following the joint technical session at The Inn, Council mem-

bers of the A.S.H.V.E. and *A.S.R.E.* met for luncheon in the main dining room. Those present included W. H. Carrier, Newark, N. J., past president of both societies, and Pres. L. S. Morse, 1st Vice-Pres. H. M. Williams, 2nd Vice-Pres. Crosby Field, A. H. Baer, C. F. Holske, G. E. Hulse, C. R. Logan, Glenn Muffly, D. E. Perham, B. E. Seamon, J. L. Shrode, F. C. Stewart, and L. A. Tucker of the *A.S.R.E.;* Pres. G. L. Larson, 1st Vice-Pres. D. S. Boyden, 2nd Vice-Pres. E. H. Gurney, Treas. A. J. Offner, R. C. Bolsinger, S. H. Downs, W. L. Fleisher, Prof. C. M. Humphreys, Prof. F. E. Giesecke, J. F. McIntire, M. C. Beman, John Howatt, and W. A. Russell of the A.S.H.V.E.

Eichberg Memorial Cup

During the Semi-Annual Banquet, W. P. Culbert, chairman of the Golf Committee, spoke of one of the most popular Philadelphia members, whose death occurred during the past year, and said that Philadelphia Chapter had offered a cup in memory of W. Roy Eichberg. The cup is to be played for at Semi-Annual Meetings of the Society by members of Chapters, the award to be made on the basis of the three lowest scores turned in by members of one Chapter.

The final session of the three-day meeting of the Society at Buck Hill Falls was opened by President Larson on June 24.

Resolutions

At the conclusion of the technical program, W. H. Driscoll presented the following resolutions, which were unanimously adopted.

BE IT RESOLVED, that the thanks and appreciation of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS assembled in Semi-Annual Meeting be expressed:

To the Philadelphia Chapter, for the friendly hospitality extended by each of its members and more especially by the Committee on Arrangements, whose efforts have again resulted in a never to be forgotten meeting of the Society.

To the Management of The Inn and its staff, whose cooperation and interest have added to the comfort and enjoyment of members and guests.

To the Daily Press and the Trade Papers, for the publicity given this Semi-Annual Meeting and the work of the Society and for the attendance of individual members.

To the Authors of Papers, in appreciation of the time spent in the preparation and presentation of important data on subjects of current interest.

PROGRAM SEMI-ANNUAL MEETING, 1936

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

THE INN, BUCK HILL FALLS, PA.

JUNE 22-24, 1936

(All events scheduled on Daylight Saving Time)

Sunday, June 21

7:30 P.M. Council Meeting (Room 233)

Monday, June 22

Technical

8:30 A.M. Registration

9:30 A.M. Greeting

Amendments to Constitution and By-Laws, R. H. Carpenter, *Chairman*
Technical Papers—

Corrosion Studies in Steam Heating Systems, by R. R. Seeber, F. A. Rohrman and G. E. Smedberg

Performance of an Underfeed Domestic Stoker, by T. G. Estep and D. C. Saylor

Relative Absorption of the Various Sections of a Round Heating Boiler, by A. J. Johnson and P. A. Mulcey

The Distribution of Steam in Heat Transfer Surface, by John McElgin

12:15 P.M. Meeting of Committee on Research (Luncheon—Main Dining Room, The Inn; Meeting—Room 233)

Entertainment

(FOR THE MEN)

1:30 P.M. Men's Golf Tournament—18 hole Kicker's Handicap for Members and Guests—Buck Hill Falls Course

Bowling on the Green

Tennis

Trout Fishing Party

6:00 P.M. Leave for Skytop Mountain

7:00 P.M. Picnic and Steak Fry on Skytop Mountain with A. S. R. E. Members

(FOR THE LADIES)

9:00 A.M. Registration

12:30 to 1:30 P.M. Luncheon—Main Dining Room, The Inn

2:00 P.M. Swimming Party at the Pool

4:00 P.M. Reception and Tea in Honor of Mrs. G. L. Larson

6:00 P.M. Leave for Skytop Mountain

7:00 P.M. Picnic and Steak Fry on Skytop Mountain with A. S. R. E. Members

Tuesday, June 23

Technical

8:30 A.M. Breakfast Meeting of Nominating Committee (Bluestone Room, Lower Lobby, The Inn; Meeting—Room 233)

9:30 A.M. Joint Session of *A. S. R. E.* and A. S. H. V. E., W. H. Carrier presiding:

Theory and Practice of the Heat Pump, by C. W. Chamberlain

Application Factors Which Govern the Selection of Refrigerating Equipment for Air Conditioning Service, by J. R. Hertzler

Water Supply as Affected by the Demand for Summer Air Conditioning, by D. C. Morrow

Progress in Air Conditioning in the Last Quarter Century, by W. H. Carrier

12:15 P.M. Joint Luncheon—Councils of A. S. H. V. E. and *A. S. R. E.* (Main Dining Room, The Inn—Buck Hill Falls)

Meeting of Guide Publication Committee (Luncheon—Main Dining Room, The Inn; Meeting—Room 233)

Entertainment

(FOR THE MEN)

1:30 P.M. Men's Golf—Buck Hill Falls Course—18 hole Medal Play Tournament for Members and Guests—Research Cup Tournament and Philadelphia Chapter Cup Tournament

Sightseeing Trip Through Poconos

Horseback Riding

Outdoor Shuffle Board

7:45 P.M. Banquet and Dance—Singing by Adelphia Quartet and Music by Adelphia Orchestra

Group Singing led by Christie—Accordionist

(FOR THE LADIES)

9:00 A.M. Registration

10:00 A.M. Sightseeing Trip Through Poconos

Ladies' Golf Tournament—18 hole Medal Play—Buck Hill Falls Course

Swimming Party at the Pool

12:30 to 1:30 P.M. Luncheon and Bridge Party—Buck Hill Tennis Club

7:45. P.M. Banquet and Dance—The Inn

Wednesday, June 24

Technical

9:30 A.M. Technical Papers—

Development of Testing Apparatus for Thermostats, by D. D. Wile

Pyrheliometers and the Measurement of Total Solar Radiation, by L. A. Harding

A Field Study of the Heat Requirements of a College Building, by F. E. Giesecke and W. H. Badgett

Heating Requirements of an Office Building as Affected by Weather Conditions, by F. C. Houghten and Carl Gutberlet

Entertainment

(FOR THE MEN)

2:00 P.M. Optional Golf at Skytop or Buck Hill Falls Course

Trout Fishing Party

Bowling on the Green

(FOR THE LADIES)

9:00 A.M. Ladies' Golf

Bowling on the Green

Swimming Party at the Pool

12:30 to 1:30 P.M. Luncheon—Main Dining Room—The Inn

2:00 P.M. Outdoor Games

COMMITTEE ON ARRANGEMENTS

R. C. BOLSINGER, *General Chairman*
W. F. SMITH, *Vice-Chairman*

H. G. BLACK .... *Registration*
J. H. HUCKER .... *Entertainment*
W. P. CULBERT .... *Golf*
M. F. BLANKIN .... *Banquet*
M. C. GILLETT .... *Ladies*
L. P. HYNES .... *Finance*
H. H. ERICKSON .... *Publicity and Transportation*
MISS JANE LOUISE BOLSINGER .... *Hostess*

No. 1037

CORROSION STUDIES IN STEAM HEATING SYSTEMS

By R. R. SEEBER * (*MEMBER*), F. A. ROHRMAN,** AND G. E. SMEDBERG *** (*NON-MEMBERS*), HOUGHTON, MICH.

This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in co-operation with the Michigan College of Mining and Technology

THIS paper is the second of a series presenting information obtained in the investigation of corrosion in steam heating systems.[1]

Up to the present time 40 corrosion tests have been conducted at various pressures and with various gas concentrations. In them the following points were considered:

1. The effect of pressure on the corrosion rate.
2. The effect of oxygen and carbon-dioxide on the corrosion rate.
3. Correlation of the rate of corrosion, as measured by the corrosion tester, to the actual life of pipe.
4. An electric resistance method of determining corrosion rates.

The results of these tests are offered now in order to make the information available to those interested and thereby to promote discussion. However, many additional tests are necessary before the role that each factor plays in the corrosion problem is fully understood.

The authors wish to stress the effect of change in certain operating conditions upon the rate of corrosion and to call attention to a method of determining rates of corrosion which seems to offer the possibility of greatly reducing the time required for such tests.

DESCRIPTION OF APPARATUS USED

The corrosion testing apparatus is essentially a closed and wet-return type of heating system which is similar in construction to the ordinary steam house-

* Professor of Mechanical Engineering.
** Assistant Professor of Chemistry.
*** Research Mechanical Engineer.

[1] Corrosion Studies in Steam Heating Systems, by R. R. Seeber, F. A. Rohrman and G. E. Smedberg, A. S. H. V. E. TRANSACTIONS, Vol. 40, 1934.

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, June, 1936, Buck Hill Falls, Pa., by R. R. Seeber.

heating system, consisting of steel piping and cast iron radiators. It uses a 66-gal boiler for generating the necessary steam. In the boiler are inserted four immersion-type heaters with a total capacity of 9 kw. The pressure of the system is kept constant by proper controls. From the boiler, steam flows through four corrosion loops in which the elements for the determination of the rate of corrosion are placed. At the bottom of each corrosion loop, valves are located to draw off the samples of condensate for analysis of oxygen and carbon-dioxide content and for pH values. A 40 sq ft radiator is in the line midway between the second and third corrosion loops. The condensate and the steam from the last loop flow into a 16-gal receiver and thence to the boiler. To cut down infiltration of the atmospheric gases, all valves above the water line of the system are packless and all joints are tight. A Beach-Russ vacuum pump is used to maintain a vacuum when desired.

Distilled water is added to the boiler by means of a connection on the top of the boiler; city water is added through the feeder line.

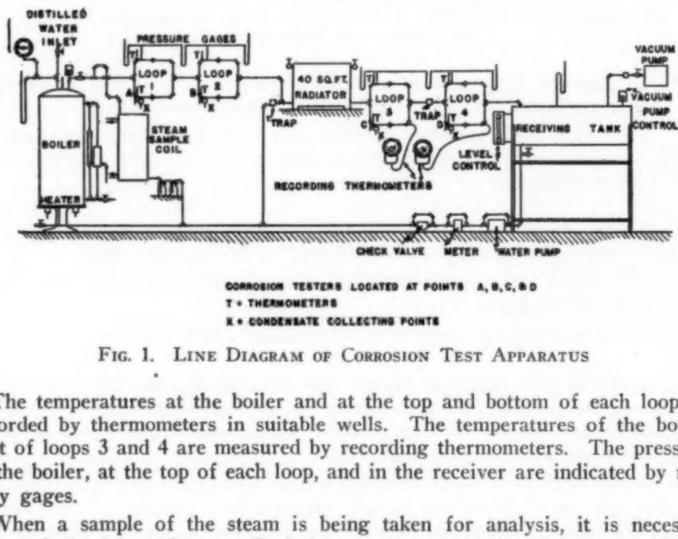


FIG. 1. LINE DIAGRAM OF CORROSION TEST APPARATUS

The temperatures at the boiler and at the top and bottom of each loop are recorded by thermometers in suitable wells. The temperatures of the bottom part of loops 3 and 4 are measured by recording thermometers. The pressures in the boiler, at the top of each loop, and in the receiver are indicated by mercury gages.

When a sample of the steam is being taken for analysis, it is necessary to entrain in the condensate all of the gases accompanying the steam. This is done by the use of a water-cooled condensation coil (Fig. 4).

METHODS OF WITHDRAWING CONDENSATE SAMPLES

One of the most difficult procedures encountered is that of withdrawing from the apparatus a sample of condensate accurately representative of the conditions inside the system. In the course of the first eleven tests four different methods were tried before satisfactory ones were found. They were:

1. Nitrogen method (Fig. 5).
2. Bottle-train method (Fig. 6).
3. Mercury-bottle method (Fig. 7).
4. Water-syphon method (Fig. 7).

In the first method a stream of nitrogen was used to expel the air from a vertical pipe section attached to the loop. The condensate was then collected in this vertical section and forced out into the collecting bottles by means of a stream of nitrogen. This method, though preventing the pollution of the condensate by the air, gave a lower concentration of oxygen and carbon-dioxide

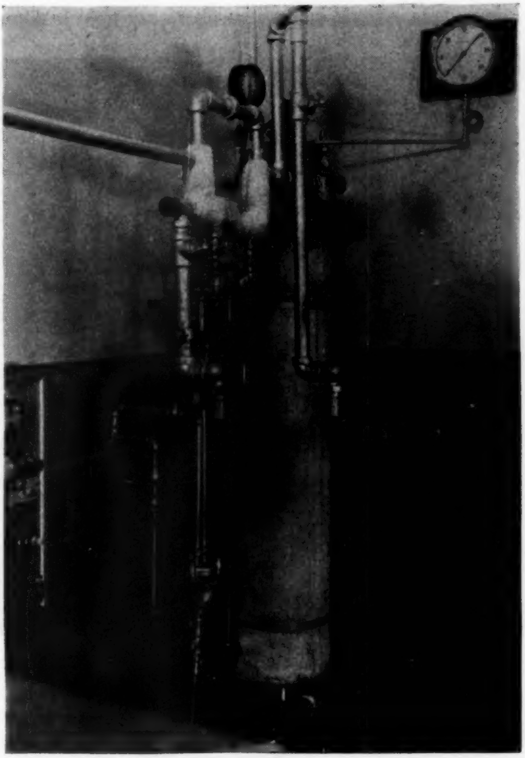


Fig. 2. View of Boiler

because of the high partial pressure of the nitrogen, and was therefore discarded.

In the second method the condensate was permitted to flow through a train composed of two bottles which were connected in series below the loop. With this method it was possible for the air originally in the bottles to contaminate the sample of condensate. To overcome this objection a third method was devised which proved successful and is being used. In it the air in the bottles is expelled by means of mercury, and as the mercury is withdrawn the condensate flows in to replace it. Apparently this method gives a sample of con-

densate which is as representative of actual conditions as can be obtained. The fourth method consisted of the use of the same bottles as in the third method, with a vacuum produced inside the train by the use of a water syphon. After a vacuum was obtained inside the train, the valve from the loop was turned on and the train filled with condensate. This fourth method, like the third, appeared to be entirely acceptable and is being used in the work.

Washed asbestos, glass wool, and cotton filters are used in filtering the con-

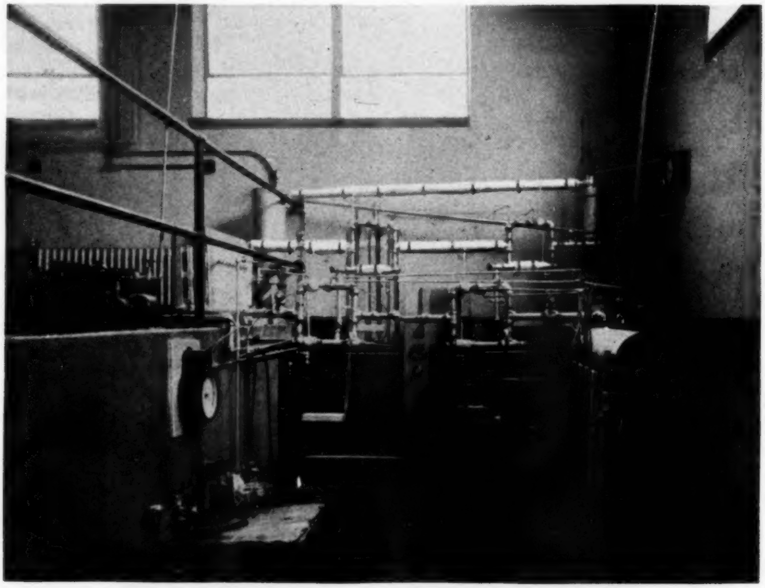


FIG. 3. VIEW OF APPARATUS

densate sample as it is withdrawn from the loop. Cotton filters seem to operate with the least difficulty.

DESCRIPTION OF TESTER AND METHOD OF CLEANING

The corrosion tester used in this work is of the type devised by the *National District Heating Association* (Fig. 8). It consists of a standard ¾-in. pipe plug, into which is threaded a brass retaining frame. The corrosion specimen consists of three helical coils mounted in the frame and insulated from one another and from the supporting frame by micarta couplings. The terminal micarta couplings are threaded into the frame and the pipe plug.

The coils are made of Bessemer steel wire 0.05 in. in diameter; the steel had been taken from a heavily cropped ingot (below the upper one-third), and pickled and annealed. The coils are approximately 11/32 in. in outside diameter

and 1 in. in length, and weigh approximately 3 grams. Except for the coils, all metal surfaces exposed to the corrosive action are protected by Sarva paint.

The accompanying table shows the analysis of the wire used in making the coils.[2]

Phosphorus	0.10 per cent
Sulphur	0.07 per cent—0.08 per cent
Manganese	0.04 per cent—0.10 per cent
Carbon	0.08 per cent—0.10 per cent

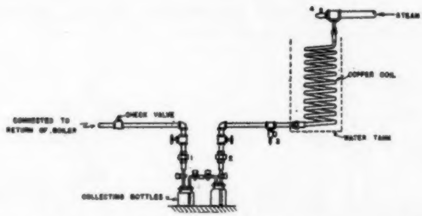


FIG. 4. STEAM SAMPLE COIL AND COLLECTING BOTTLES

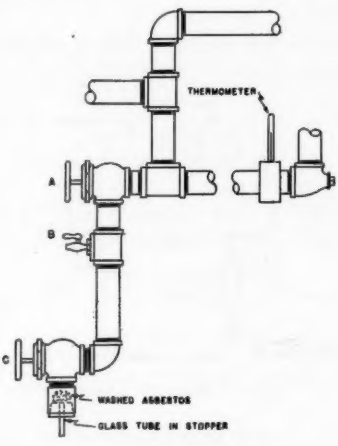


FIG. 5. NITROGEN METHOD

During the progress of the work many methods were tried to clean the wire samples before use. Two methods have been particularly successful. The first, suggested by the *National District Heating Association,* is given.

The coils are placed in a flask, containing ether or carbon tetrachloride and equipped with a reflux condenser, and are boiled for approximately 30 min.

[2] Analysis by American Steel and Wire Co.

The coils are then removed from the solvent and dried at a constant temperature of 107 F for two hours.

The second method consists of immersing the coils in a solution of dilute sulphuric acid and drying them with benzol or acetone.

Of several methods of cleaning the samples after use, it was found that the most satisfactory is to boil the coils in a solution of sodium hydroxide and zinc and dry them with pure benzol. Other methods which were tried included the following:

1. Cleaning the coils by passing an electric current through a slightly acid solution, the wire samples being used as the cathode.

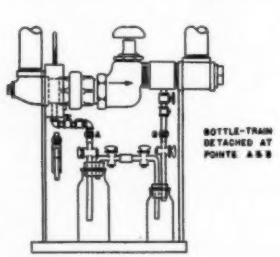


FIG. 6. BOTTLE-TRAIN METHOD

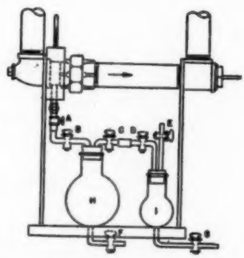


FIG. 7. MERCURY-BOTTLE METHOD

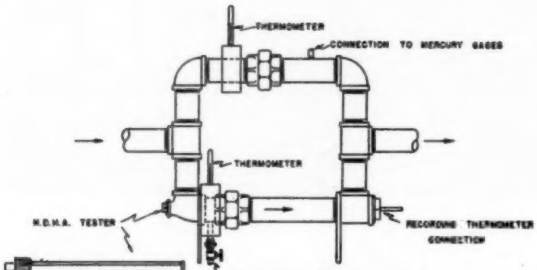


FIG. 8. CORROSION TEST LOOP

2. Immersing the coils in a 0.33 per cent solution of sulphuric acid for a short period of time.

3. Immersing the coils in a 15 per cent solution of sulphuric acid with 5 per cent of the inhibitor. (Super-In-Control.)

METHOD OF PROCEDURE IN TEST

At the beginning of each test a new set of wire coils are cleaned, weighed, and placed upon the insulating frames. Water is added to the boiler, together with the amount of chemicals needed to produce the required type of steam, and the apparatus is started. After the steam has been tested and found to be

as desired, the corrosion testers are placed in their respective loops and new charts are placed in the recording thermometers. The system is then operated for approximately 200 hours.

The first condensate samples are withdrawn for analysis the day after the test is started, and the second and final samples the day before the test is completed. Usually, also, both the condensate and the steam are analyzed the day the test is ended.

The gages are read each time an analysis is made. After the test has been conducted for the required length of time, the apparatus is shut down and the corrosion coils are removed from the loops, cleaned of their products of corrosion, dried, and weighed. From the loss in weight the corrosion rate in inches penetration per year is calculated by the following formula:

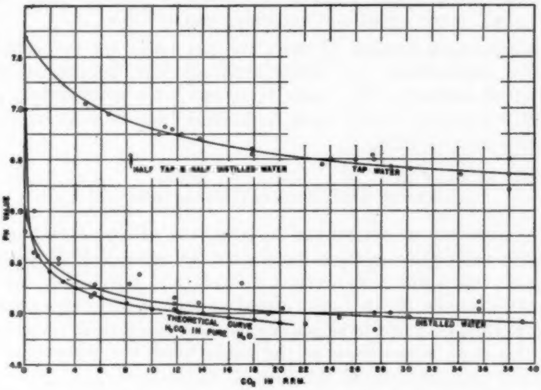


FIG. 9. CARBON DIOXIDE-pH CURVE

$$R = \frac{A}{B \times C}$$

where:

R = corrosion rate as average penetration in inches per year.
A = loss of weight in pounds per square inch.
B = weight of metal in pounds per cubic inch = (0.2782 lb).
C = duration of the test in years.

METHOD OF ANALYSIS

The reagents and the methods of analysis used in this work are those recommended by the *National District Heating Association,* with certain additions and variations.

The samples, after being withdrawn from the test set-up, are placed in a cooling vat and cooled to 20 C; they are then analyzed for oxygen, carbon dioxide, and pH value.

The Winkler method of oxygen analysis is used throughout.

Three methods of carbon dioxide analysis were tried, two of which were

discarded. The first method consisted of adding to 100 cc of the condensate sample 25 cc of barium hydroxide, and titrating with HCl to the phenolphthalein end-point. This method of analysis, it was found, introduces the possibility of considerable error unless enough barium chloride is added to reduce greatly the solubility of barium carbonate. Further, in it the time element presents a difficulty: if the amount of HCl necessary to titrate the solution is added and a period of 5 to 10 min is allowed to elapse before more HCl is added, the results obtained by this method do not check those obtained by the other methods.

The second method was by direct titration with sodium carbonate.

The third method was by direct titration with barium hydroxide (Dr. Guernsey) to the phenolphthalein end-point. ($Ba(HCO_3)_2$). This method was found to be much more rapid and accurate, and is being used.

During the greatest number of tests the pH value of the condensate was measured by a quinhydrone set, where accuracy was determined by checking it against that of another of the same type, against a comparator, and against standard buffer solution. Later tests have made use of a glass electrode for measurement of the pH value.

Carbon Dioxide—pH Relationship

To test the quinhydrone set-up and to study the effect of carbon dioxide on the pH value, a series of tests was conducted by saturating different solutions of distilled and tap water with carbon dioxide and measuring the pH value. (Fig. 9 shows a curve for pH as related to carbon dioxide content for tap water and for distilled water and also shows the theoretical curve of the relationship.) The close comparison of the actual with the theoretical values appears to confirm the accuracy of the method of analysis.

Time Required for Tests

In order to determine the correct length of time for each test, the time taken for the first eleven tests was varied from one to four weeks. A period of about 200 hours was found to be the most desirable.

Pressure vs. Corrosion Rate

The effect of pressure upon the corrosion rate was studied during the first eleven tests by using the same quality of steam but varying the pressure from 20 in. Hg vacuum to 2 lb per square inch gage.

Discussion of Results

The corrosion rates in loops 1 and 2 on the supply side of the radiator were practically zero for every condition of operation. This result was expected, as it is found in practice that supply lines are not much corroded. Since this is the case, the discussion concerns itself with only the loops 3 and 4 on the return side of the system and may be divided as follows:

1. Relation of pressure to corrosion rate.
2. Relation of rate of condensation to corrosion rate.

3. Relation of temperature to corrosion rate.
4. Correlation tests.
5. Electric resistance method of determining corrosion rates.

1. In the tests the steam qualities were kept as nearly constant as possible, while the system operated at various pressures ranging from 20 in. Hg vacuum to 2 lb per square inch gage. Previous to these tests the system was made as tight as possible by several applications of shellac to all the joints of the system while under vacuum. Table 1 gives the data of the first eleven tests

TABLE 1. DATA AS BASIS FOR CURVE 10, WITH STEAM QUALITIES CONSTANT

Test No.	Loop No.	Gage Pressure Lb/Sq In.	Average Temp. F	Average pH	Corrosion Rate In. Pent./Yr × 1000			
					A	B	C	Average
1	3	2 lb	109	8.3	9.63	12.41	9.13	10.39
1	4	2 lb	93	8.0	10.29	10.82	10.32	10.48
2	3	2 lb	109	7.4	10.13	11.36	8.97	10.15
2	4	2 lb	94	7.0	10.55	7.28	10.50	9.44
3	3	20 in. Hg	110	8.2	1.20	0.475	1.82	1.20
3	4	20 in. Hg	110	8.2	6.19	8.62	7.11	7.30
4	3	20 in. Hg	101	8.0	6.17	2.94	3.08	4.06
4	4	20 in. Hg	93	7.7	13.02	11.16	9.00	11.07
5	3	0 lb	101	7.3	11.93	14.00	15.00	13.64
5	4	0 lb	88	7.1	6.31	5.72	7.60	6.54
6	3	0 lb	157	8.3	1.13	0.75	0.89	0.92
6	4	0 lb	140	7.7	3.67	2.57	3.53	3.25
7	3	0 lb	83	6.02	8.34	9.54	7.71	8.53
7	4	0 lb	82	5.9	8.39	6.24	4.99	6.52
8	3	1 lb	87	5.8	10.44	10.43	7.69	9.52
8	4	1 lb	83	5.8	7.30	8.52	5.70	7.17
9	3	1 lb	115	5.9	12.06	9.57	10.80	10.81
9	4	1 lb	101	5.8	14.24	9.72	10.79	11.58
10	3	10 in. Hg	120	6.9	6.10	7.32	6.37	6.77
10	4	10 in. Hg	115	6.7	9.07	7.75	6.98	8.27
11	3	10 in. Hg	137	7.0	5.40	4.56	4.06	5.00
11	4	10 in. Hg	134	7.0	7.57	5.29	5.39	6.08

used as the basis for the curve shown in Fig. 10. This curve gives the relationship of the corrosion rate in loop No. 3 to the operating pressure, several factors influencing this relationship. At the higher pressures the gases are forced to stay in solution and the rate of flow of condensate is greater, both conditions acting to increase the corrosion rate. Under vacuum operation the gases tend to be drawn from the solution and the rate of flow is lower.

2. The rate of flow of condensate has an important bearing on the corrosion rate. Fig. 11 shows the rate of condensation in the various loops of the system under different pressure conditions. It is difficult to compare the corrosion rate as obtained from this apparatus with that in a large commercial installation, because of the small rate of flow of condensate. The higher velocity tends to wash off both the protective coatings which form and also the products of corrosion which protect the metal to some extent. There is a possibility of some erosive action in cases of high velocity.

3. The temperature of the corroding medium is an important factor, as brought out by Speller and others. A series of tests was conducted with various temperatures in loop 3. Figs. 12 and 13 show how the corrosion rate varied with the temperature. In loop 3 the corrosion rate increased with the temperature up to a certain point and then dropped as the temperature in-

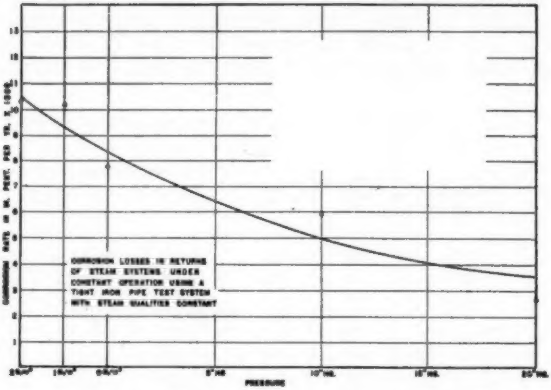


FIG. 10. RETURN LINE PRESSURE CORROSION RATE CURVE

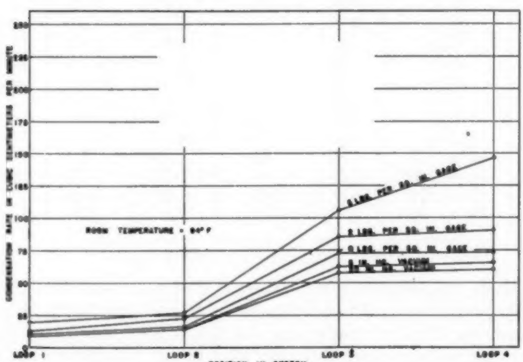


FIG. 11. CONDENSATION RATES OF STEAM APPARATUS

creased. This action is due to the fact that the increase in temperature drives the gases from the solution into the gaseous phase above.

The curve in Fig. 13, taken from Speller's *Corrosion Causes and Prevention*,[3] shows the effect of temperature in both an open and a closed system. The presence or absence of a gaseous phase above the corroding solution has a marked effect on the rate of corrosion as related to temperature. In the

[3] 1st edition, 1926.

arrangement used in this work a gaseous phase was present in the top of each loop; consequently the gases were able to escape and thus to cause a drop in the corrosion rate.

4. To relate the rate of corrosion as measured by the *National District Heat-*

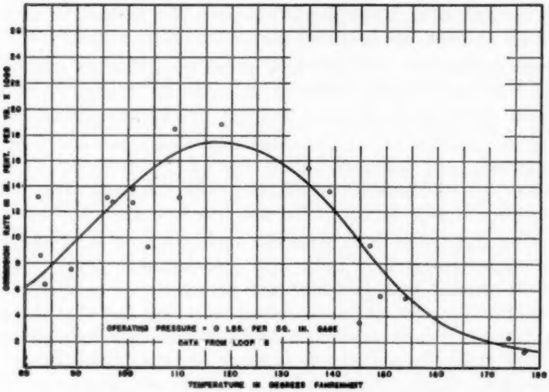


FIG. 12. CORROSION RATE VERSUS TEMPERATURE

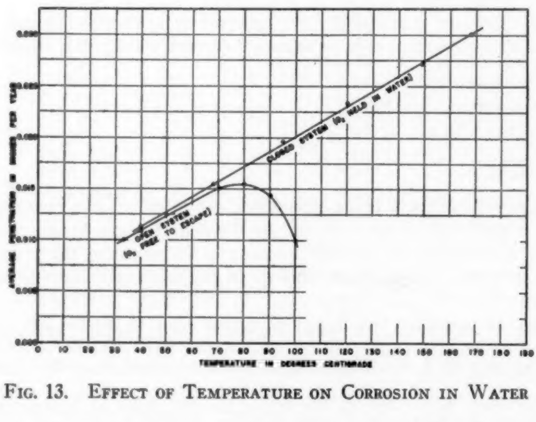


FIG. 13. EFFECT OF TEMPERATURE ON CORROSION IN WATER

ing Association Tester, which is given in inches of penetration per year, with the actual life of pipe, these tests were conducted as follows:

In the college heating system a pipe loop was inserted in a return line. In this loop was mounted a standard corrosion tester and a section of black iron pipe in which had been milled three spots 0.003 in. thick. Fig. 14 shows the construction.

The wire coils were weighed and placed on the retaining frame, which was then installed in the loop. Water was passed through the tester until one of

the thin sections broke through, the test coils being removed and their corrosion rate determined. From the corrosion rate and the time in breaking through the thin section the correlation was made.

Each individual test requires about six months to complete, and several tests are necessary to arrive at an accurate average of the relationship. The tests so far have been made at atmospheric pressure, the inside of the thin section being turned to a smooth surface and the thin sections milled.

The average penetration rate of these three tests roughly checks that indicated by the *National District Heating Association* testing wires, the figures being 100 units penetration by break-through to 84 units shown by the tester.

5. The use of the increase of the resistance of a specimen to determine its corrosion penetration is not new. Early use of this method was made by E. Wilson.[4] Later, J. C. Hudson[5] made tests with the method to determine its accuracy, and used it extensively in his study of atmospheric corrosion.

The following discussion deals with the application of the resistance method of corrosion measurement in a tight iron pipe steam system. The method is based upon the proved principle that the resistance of a uniform conductor varies inversely as the area of its cross-section. The fact that the products of corrosion have such a high resistance as compared with the resistance of the conductor itself makes it possible to measure the true resistance of the metal itself without first cleaning it. This has a distinct advantage over the gravitational method, which requires that the specimen be thoroughly cleaned without removing the metal itself.

In order to measure successfully the amount of corrosion by the change of the resistance of a specimen it is necessary to overcome several obstacles:

1. Contact resistance.
2. Type of specimen and holder.
3. Temperature effects.

Contact Resistance

When low resistances are being measured, the effect of the contact resistance can best be eliminated by the use of the Double Kelvin bridge circuit. This circuit (Fig. 14) makes it possible to accurately measure the true resistance between the potential leads with only a negligible error due to the contact resistance.

The bridge is operated by first adjusting the branch resistors so that $\frac{R_a}{R_b}=\frac{R_c}{R_d}$. The standard resistance, R, is then adjusted until the galvanometer shows no deflection upon closing the key. In this state of balance the current flowing through the unknown resistance, X, is necessarily the same as that flowing through R. The relation between the standard resistance and the unknown resistance is then determined by the four branch resistors—*i.e.*,

$$\frac{R}{X}=\frac{R_b}{R_a}=\frac{R_d}{R_c}$$ [6]

[4] E. Wilson, *Proc. Physical Soc.*, 1926, 39, 15.
[5] J. C. Hudson, *Proc. Physical Society*, 1928, 40, 107.
J. C. Hudson, *Trans. Faraday Soc.*, 1929, 25, 48.
[6] A detailed description of the Kelvin Bridge can be found in Notes on the Kelvin Bridge. Note Book No. 4, by the Leeds and Northrup Co.

Type of Specimen and Holder

The ideal specimen to use for this system must have necessary strength, proper steel composition, uniformity, and large exposed area as compared with volume. Further, it must cause a minimum of errors due to pitting and it must secure proper connection of potential leads without introducing a different kind of metal. It was found that a thin ribbon of steel would fulfill these requirements most fully, as shown in the following discussion.

If a conductor of circular cross-section is used, the diameter must be reduced in order to increase the ratio between the exposed area and the volume. This

TABLE 2. CORROSION TEST DATA WITH STEAM QUALITIES CONSTANT

TEST No.	LOOP No.	GAGE PRESSURE LB/SQ IN.	AVERAGE TEMP. F	AVERAGE pH	CORROSION RATE IN. PENT./YR × 1000			
					A	B	C	Average
1	3	2 lb	109	8.3	9.63	12.41	9.13	10.39
1	4	2 lb	93	8.0	10.29	10.82	10.32	10.48
2	3	2 lb	109	7.4	10.13	11.36	8.97	10.15
2	4	2 lb	94	7.0	10.55	7.28	10.50	9.44
3	3	20 in. Hg	110	8.2	1.20	0.475	1.82	1.20
3	4	20 in. Hg	110	8.2	6.19	8.62	7.11	7.30
4	3	20 in. Hg	101	8.0	6.17	2.94	3.08	4.06
4	4	20 in. Hg	93	7.7	13.02	11.16	9.00	11.07
5	3	0 lb	101	7.3	11.93	14.00	15.00	13.64
5	4	0 lb	88	7.1	6.31	5.72	7.60	6.54
6	3	0 lb	157	8.3	1.13	0.75	0.89	0.92
6	4	0 lb	140	7.7	3.67	2.57	3.53	3.25
7	3	0 lb	83	6.0	8.34	9.54	7.71	8.53
7	4	0 lb	82	5.9	8.39	6.24	4.99	6.52
8	3	1 lb	87	5.8	10.44	10.43	7.69	9.52
8	4	1 lb	83	5.8	7.30	8.52	5.70	7.17
9	3	1 lb	115	5.9	12.06	9.57	10.80	10.81
9	4	1 lb	101	5.8	14.24	9.72	10.79	11.58
10	3	10 in. Hg	120	6.9	6.10	7.32	6.37	6.77
10	4	10 in. Hg	115	6.7	9.07	7.75	6.98	8.27
11	3	10 in. Hg	137	7.0	5.40	4.56	4.06	5.00
11	4	10 in. Hg	134	7.0	7.57	5.29	5.39	6.08

seriously reduces its strength and allows large pitting errors. However, by reducing its thickness a conductor of rectangular cross-section can be given a larger ratio of exposed area to volume without loss of the necessary strength. This shape also helps to eliminate the error due to pitting since a pit affects only a much smaller portion of the cross-section. Furthermore, potential leads can be formed by splitting the ends of the specimen. Thus both leads can be brought out of the corrosive medium before any artificial joints are made.

The preparation of the specimen is important because of the necessity of a proper steel composition and of uniformity of cross-section. Rings about ¾ in. in width are cut from a standard 2 in. iron pipe and split open. These are hot rolled into strips approximately 18 in. long by ¾ in. wide by 0.015 in. thick. The strips are then ground down as thin as is practical. By careful machining the specimen is made as uniform as possible, a point of importance when the resistance changes are being reduced to penetration units. Finally, the finished strips are thoroughly annealed without oxidizing.

The specimen holder is so designed that the resistance readings can be taken without disturbing the system or removing the specimen. The current and potential leads are taken out through a split micarta plug. The drawings in Fig. 16 show clearly the method of bringing out the leads and the arrangements made for inserting insulated thermocouples

Temperature Effects

With the use of an iron conductor and the Kelvin Bridge, temperature has two effects: the first is to change the resistance of the specimen as the temperature changes; and the second is to set up an unbalanced thermocouple emf

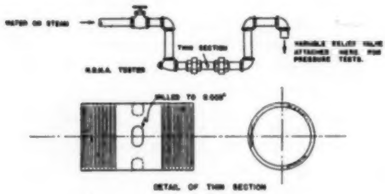


FIG. 14. CORRELATION TESTER

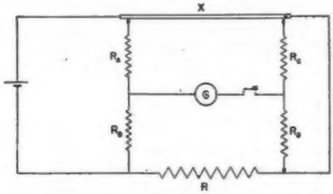


FIG. 15. DOUBLE KELVIN BRIDGE

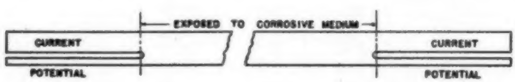
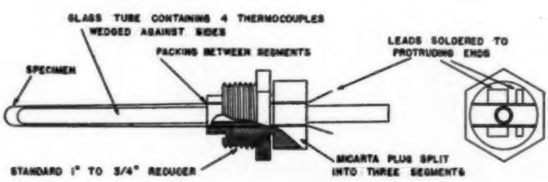


FIG. 16. (ABOVE) SPECIMEN. (BELOW) HOLDER



when the two iron and copper junctions of the potential leads are at different temperatures. The latter effect is easily eliminated by placing the two junctions in an ice-filled vacuum bottle so that they will remain at a constant and equal temperature.

The resistance of the iron specimen used does not increase directly as the temperature rises, but begins to increase more rapidly at the higher temperatures. One way to correct for temperature changes is to take the temperature of the specimen at the time of the resistance reading and multiply this resistance as read by the proper correction factor as taken from a prepared table. This necessitates the use of an expensive potentiometer to get temperature readings correct to the nearest tenth of a degree Fahrenheit. Several additional chances for error also enter the measurements, not to mention the extra work entailed. A more effective method is described.

On the same sheet with the resistance vs temperature curve is also plotted the emf vs temperature curve of a copper and constantan thermocouple. See Fig. 17, which shows that for the range of temperatures plotted the specimen resistance and thermocouple voltage bear a constant relationship to each other. The fact that these curves follow each other so closely is utilized in this method. As shown in the circuit diagram, Fig. 18A, the thermocouple is combined directly with the bridge circuit by being placed in series with the galvanometer. The sensitive junction of the thermocouple is placed in the corrosive solution near the specimen, but is highly insulated from the solution and specimen. The arrangement and method of insulating are shown in Fig. 16.

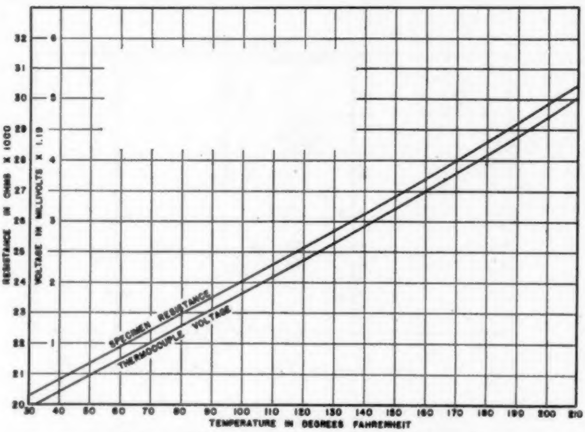


FIG. 17. SPECIMEN RESISTANCE AND THERMOCOUPLE VOLTAGE

Using the letters on the diagram, Fig. 18A, the mathematical solution is:

Let

I = Current through specimen in milliamperes.

r = Resistance of each branch of the bridge.

R = Resistance of specimen at the temperature of cold junction zone.

$\triangle R$ = Increase in resistance of specimen because of temperature increase above that of cold junction zone.

MV = emf furnished by thermocouple in millivolts.

Assume that the circuit is balanced with the standard resistance at R, that the temperature of the specimen is above that of the cold junction zone, and that no current is flowing through the galvanometer. The equations can be obtained from this circuit,

$$I\,(R + \triangle R) + i_2 r - MV - i_1 r = 0 \qquad (1)$$

$$IR + i_2 r + MV - i_1 r = 0. \qquad (2)$$

By subtracting (2) from (1) the equation is obtained,

$$I \triangle R = 2\,MV.$$

It is now apparent that by proper control of the current through the specimen, the resistance as read will have been automatically corrected to the resistance at the temperature of the cold junction zone. It is also apparent that as the cold-junction resistance of the specimen changes, the current adjustment also must be changed in order to bring about the proper temperature corrections. The accuracy with which the current must be adjusted depends on the temperature difference between the cold junction and the specimen. If this is about 100 F and the proper current is about 3 amperes, the current must be adjusted correctly to the third decimal (3.000) in order to assure a temperature correction to the nearest tenth of a degree.

The current is adjusted with the necessary accuracy as shown in Fig. 18B.

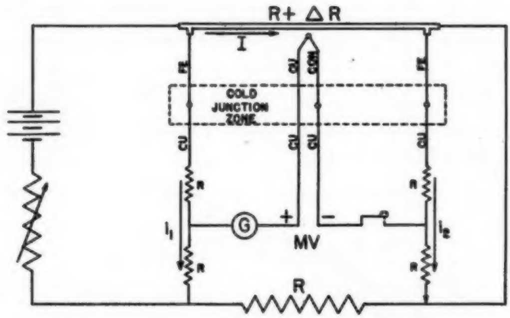


FIG. 18A. DIAGRAM OF THERMOCOUPLE CONNECTION

FIG. 18B. CURRENT ADJUSTMENT

The current in passing through the manganin slide wire, *AB*, produces a voltage drop which is balanced against the standard cell. Since the electromotive force of the standard cell and the resistance of the slide wire are constant at room temperatures, the scale can be calibrated in amperes. However, since the current adjustment is dependent upon the resistance of the specimen, it is most convenient to mark the scale directly in ohms corresponding to the cold junction resistance of the specimen.

CONCLUSION

A schematic diagram of the complete circuit is shown in Fig. 19. Four thermocouples are used in series, an arrangement which gives a more sensitive adjustment and a better average temperature correction. The galvanometer and key are connected to the blades of a D. P. D. T. switch, so that they can be conveniently used for either current or resistance adjustments. All re-

TABLE 3. CORROSION TEST DATA

TEST No.	LOOP No.	GAGE PRESS.	AVE. O_2 CC/1	AVE. CO_2 PPM	AVE. TEMP. F	AVE. pH	CORROSION RATE IN. PENT./YR × 1000			
							A	B	C	Ave.
12	3	0	1.9	41	84	5.9	7.83	5.72	5.32	6.29
12	4	0	2.0	48	79	5.1	5.57	4.59	7.91	6.02
13	3	0	1.2	35	89.3	5.8	7.89	6.84	7.67	7.47
13	4	0	1.0	36	81.7	5.8	5.13	4.96	4.04	4.71
14	3	0	0.5	25	144.9	6.3	4.39	2.57	0	2.32
14	4	0	0.6	25	124.3	5.4	5.17	1.95	5.11	4.08
15	3	0	1.2	56	85	5.8	13.339	13.358	12.470	13.06
15	4	0	1.3	58	86	5.8	5.479	5.23	6.469	5.73
16	3	0	1.0	30	104	5.8	9.018	8.414	9.616	9.02
16	4	0	1.0	31	87	5.8	4.606	5.689	4.792	5.09
17	3	0	0.4	31	139	5.8	11.373	12.714	16.566	13.55
17	4	0	0.6	48	113	5.5	0.9818	0.674	0.800	0.82
18	3	0	1.1	30	108.5	5.7	17.08	17.62	20.59	18.43
18	4	0	1.3	33	93	5.7	3.56	6.54	3.90	4.67
19	3	0	0.9	24	134.4	5.9	15.14	17.31	13.21	15.32
19	4	0	1.2	28	105.5	5.9	8.04	5.15	5.45[a]	6.21
20	3	0	...	...	138	...	0.977	2.088	1.041[a]	1.37
20	4	0	...	...	109	...	2.313	1.165	0.017[a]	1.50
21	3	0	1.2	35	97	5.6	13.27	10.88	13.77	12.64
21	4	0	1.1	35	88	5.5	8.62	10.21	9.77	9.53
22	3	0	1.3	38	118	5.4	20.42	18.26	17.72	18.80
22	4	0	1.2	40	98	5.4	12.36	12.34	13.28	12.67
23	3	0	1.9	35	101	5.6	11.86	11.6	14.37	12.61
23	4	0	1.3	31	88	5.5	7.86	11.2	11.6	10.22
24	3	0	1.2	6	110	5.7	13.074	13.22	12.75	13.01
24	4	0	1.4	13	96	5.8	9.03	9.25	10.44	9.57
25	3	0	1.8	10	96	5.9	17.24	12.28	9.64	13.05
25	4	0	1.4	10	87	5.9	7.77	11.85	8.78	9.47
26	3	0	0.8	1	149	5.6	4.678	5.684	6.068	5.48
26	4	0	0.7	6	121	5.6	6.087	6.557	5.909	6.18
27	3	0	1.4	1	147	6.0	8.79	9.79	9.42	9.33
27	4	0	1.4	3	117	5.2	8.89	8.34	6.95	8.06
28	3	0	1.9	1	174	5.1	2.36	1.85	2.59	2.27
28	4	0	0.7	3	153	4.9	7.41	7.98	8.13	7.84
29	3	5 lb sq in.	1.6	0	183	5.6	1.907	2.015	2.508	2.15
29	4	5 lb sq in.	1.8	1	157	5.5	1.336	2.310	3.05	2.35
30	3	5 lb sq in.	1.1	0	192	6.2	1.66	1.75	1.66	1.69
30	4	5 lb sq in.	2.1	0	160	6.1	3.91	3.87	4.09	3.96
31	3	5 in. Hg	0.3	0	171	6.5	1.07	1.02	1.05	1.04
31	4	5 in. Hg	0.5	0	149	6.5	2.19	2.28	2.22	2.23
32	3	5 in. Hg	0.5	0	170	6.2	1.598	1.153	1.422	1.39
32	4	5 in. Hg	0.8	1	150	5.8	5.45	5.03	5.51	5.33
33	3	5 in. Hg	0.4	0	172	6.4	1.252	0.747	1.21	1.07
33	4	5 in. Hg	1.0	0	150	6.3	5.36	4.57	5.46	5.13
34	3	5 lb sq in.	0.6	0	192	6.4	1.39	1.35	2.202	1.65
34	4	5 lb sq in.	0.7	0	161	6.4	4.91	4.58	5.40	3.96
35	3	0	0.5	0	177.6	6.3	1.017	1.446	1.322	1.26
35	4	0	0.6	0	150	6.4	3.00	2.83	2.90	2.93
36	3	0	0.5	0	177	6.1	1.0666	1.2533	1.3165	1.21
36	4	0	0.6	0	147	5.9	4.3774	5.3676	3.9484	4.56
37	3	5 lb sq in.	0.5	0	190	5.8	0.9756	1.152	1.292	1.14
37	4	5 lb sq in.	0.6	5	156	5.7	4.353	5.624	5.443	5.14
38	3		No Corrosion Samples Analysis Method Tests							
38	4									
39	3	0	0.8	1	173	5.8	1.73	1.722	1.9	1.78
39	4	0	0.6	2	143	5.8	3.09	3.935	4.28	3.73
40	3	0	0.9	2	154	6.3	5.20	5.35	5.39	5.31
40	4	0	0.8	4	131	6.3	8.22	9.11	6.77	8.03

[a] Nitrogen Test.

sistors are made of manganin in order to avoid corrections due to temperature changes. Special note was made of all contacts and connections in order to reduce current variations to a minimum.

With this circuit it becomes comparatively simple to take a corrected resistance reading. Since the current scale is calibrated in ohms, it is only necessary to alternately adjust the current and standard resistance until both scales give the same reading. The current adjustment is made by turning the galvanometer switch to the position marked *I*, moving the slide *A* to the approximate resistance of the specimen, and varying the current rheostats until the galvanometer does not deflect. The resistance adjustment is now made by turning the galvanometer to the position marked *R* and adjusting the slide *B* until the galvanometer does not deflect.

With the selection of this type of specimen the calculation of the penetration

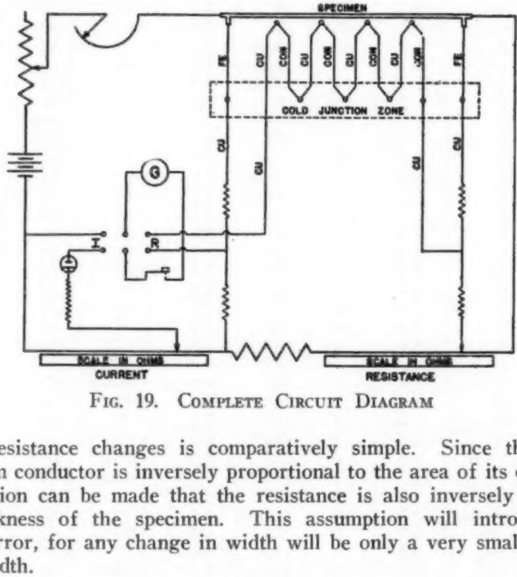


FIG. 19. COMPLETE CIRCUIT DIAGRAM

from the resistance changes is comparatively simple. Since the resistance of a uniform conductor is inversely proportional to the area of its cross-section, the assumption can be made that the resistance is also inversely proportional to the thickness of the specimen. This assumption will introduce only a negligible error, for any change in width will be only a very small per cent of the total width.

$$R_1t_1 = R_2t_2$$

Where

R_1 = original resistance.
R_2 = final resistance.
t_1 = original thickness.
t_2 = final thickness.

$$t_2 = R_1t_1/R_2.$$

The penetration is then equal to half of the total change of thickness since the specimen is corroding from two sides.

$$P = t_1 - t_2/2.$$

It is obvious that this penetration calculation can be only as accurate as the

original thickness measurements. With a good micrometer the average thickness can be estimated only to the second place. The average thickness can be measured more accurately by weighing the specimen and calculating the thickness from the length, width, and specific gravity of the strip.

The advantage of this method over the gravitational method lies chiefly in two points: (1) measurements can be made without disturbing the set-up, and (2) the increased sensitivity of the method makes it possible to get definite results in a shorter time. These two facts will facilitate the work of obtaining curves of the corrosion rate against time, temperature, and the other variables.

The range of tests conducted is not sufficiently wide as yet for an attempt to be made to correlate the various factors in their combined effect in producing corrosion in return piping. It is felt that the results given bear out the following general statements for type of system tested. Other factors being constant,

TABLE 4. STEAM SAMPLE ANALYSIS

TEST No.	LOOP No.	AVE. O_2	AVE. CO_2	TEMP. F	PH
26	3	2.3	15	70	5.9
27	3	4.5	0	70	6.4
28	3	3.8	2	70	6.1
29	3	0.3	0	70	5.6
30	3	3.9	1	70	6.2
31	3	3.2	0	70	7.1
32	3	2.7	0	70	6.9
33	3	4.4	0	70	6.5
34	3	2.0	1	70	6.6
35	3	2.1	0	70	6.8
36	3	1.1	0	70	6.6
37	3	0.9	11	70	5.45
39	3	1.4	5	70	5.8
40	3	1.5	13	70	6.6

the corrosion rate will vary directly with the oxygen content of the condensate.

Other factors being constant, the corrosion rate will vary directly with the free CO_2 content of the condensate, at least up to about 30 parts per million. One part per million of oxygen appears to have about the same corrosive effect as between twenty and thirty parts per million of CO_2.

Increase of temperature for the same pressure increases the corrosion rate up to a certain point, from where it decreases rapidly with further increase of temperature. Increased rate of flow of condensate increases corrosive effect.

Decrease of pressure carried in the system, the quality of steam being the same, decreases the corrosion rate in a tight system.

ACKNOWLEDGMENT

Acknowledgment is due to A. E. Felt, H. V. Williamson, and F. L. Partlo for assistance in the tests and in the preparation of the report.

The development of the electric resistance method and the actual construction and calibration of the instruments necessary to insure its proper operation were the work of H. V. Williamson.

DISCUSSION

R. M. PALMER (WRITTEN): This paper exhibits an adequate appreciation on the part of the authors of the varied factors involved in this important problem.

The paper is satisfactorily comprehensive. Therefore, very little can be added in the way of discussion.

I am pleased to note that the authors are interesting themselves in the role played by carbon dioxide in the problem of corrosion. Altogether too little attention has been paid to this corrosive gas in the past. This is especially true as relating to the problem of corrosion of non-ferrous metals, such as the brasses and copper.

Carbonic acid rapidly attacks both the brasses and copper and since it is almost invariably present in water, its possible effects should not be overlooked. Certain research work done abroad, notably by the *British Non-Ferrous Metals Research Association* and by some of the German research workers, has placed special emphasis on the action of carbonic acid on metals.

It has been found, for example, that the corrosion of copper, taking place when submerged in distilled water saturated with oxygen, is negligible. On the other hand, when carbon dioxide is introduced, corrosion is greatly accelerated and is progressive. Further, it has been shown that the product of corrosion of copper, when oxygen alone is present, forms an impervious, adherent protective coating, while the presence of carbon dioxide yields a product of corrosion which is pervious and non-adherent—therefore progressive corrosion takes place.

This discussion of the problem of corrosion of non-ferrous metals is not directly germane to the problem of corrosion of steam heating systems. However, it is my conviction that additional research work on the role played by carbon dioxide, in various corrosion problems encountered, will be very much worthwhile.

D. H. LITTLE[7] (WRITTEN): The authors state that "In order to determine the correct length of time for each test, the time taken for the first eleven tests was varied from one to four weeks." "A period of about 200 hours was found to be most desirable." Inasmuch as the *National District Heating Association* has published directions for using their standard corrosion testers which require that the testers remain under test for four weeks, the authors' reasons for selecting a period of 200 hours as being the most desirable would provide additional information of value.

Since some form of boiler water treatment, which is sometimes a source of carbon dioxide, is added to the feedwater in many boiler plants, an analysis of the feedwater used during each test should be included in order to permit approximate comparisons for individual cases. If no tests have been conducted where chemicals, similar to those used in commercial plants, are added to the feedwater, a series of tests of this nature would be a valuable adjunct to this paper.

The results of the corrosion tests conducted to determine the relation between the rate of corrosion and the pressure which is carried in a heating system indicate a decrease in the corrosion rate with a decrease in pressure. As the authors point out, this is accountable to the difference in solubility of dissolved gases at various pressures. However, tests which were conducted in actual building heating systems show that the corrosion rate is greater in vacuum systems than in pressure systems. Tests were conducted in systems which had not been made air tight as was the case with the test procedure in which the results are as stated in this paper. Although the curve as shown in Fig. 10 represents a true picture of the corrosion rate vs. pressure as determined under laboratory conditions, it should be emphasized that the corrosion rate in actual heating systems may be very different from what is indicated by this curve.

To date there has been a good deal of work done on general corrosion problems,

[7] Engineer, Steam Heating Service Dept., The Edison Electric Illuminating Co. of Boston.

but very little has been done along the lines of corrosion inhibitors that enter a definite chemical reaction with any of the factors of corrosion in a condensate line. As the authors of this paper state, steam supply lines are very little corroded, but a pertinent question at this time is whether or not a suitable corrosion inhibitor can be found which will be commercially practical and if it can be safely added to the steam at the boiler or whether an individual inhibiting device should be located at each customer's premises and feed only into the return lines.

Some preliminary work was conducted in Boston this year on the addition of sodium sulphite as a corrosion inhibitor in condensate return lines. The result of a series of tests conducted there showed that the addition of this chemical decreases the rate of corrosion in condensate lines inasmuch as it decreases the further oxidation of ferrous hydroxide and raises the pH value of the condensate. These tests were made using the *N.D.H.A.* standard corrosion testers and methods of analysis. As a further result of this series of tests, it has been suggested that more tests be conducted for the purpose of studying the effect of sodium sulphite, alkaline tannates, ferrous hydroxide and other reducing substances upon the rate of corrosion in steam condensate return lines.

The use of the electric resistance method of determining corrosion rates as described in the paper will serve to reduce the time element necessary to get the results as well as facilitate the work of gathering data on specimens which are located in customers' heating systems.

G. C. EATON [8] (WRITTEN): The authors of this paper, the second of this valuable series, are apparently proceeding along the lines originally laid out for this research and are to be congratulated on the results achieved.

We are particularly interested in the rapid method of determining corrosion as described. In our opinion, the apparatus developed is a very useful laboratory instrument which might possibly be used in the field, if great care were used. As the authors will no doubt agree, the analysis of steam near its source does not guarantee its purity (or impurity) at the point of use; hence field tests using this apparatus would be of distinct value in the field study of corrosion in steam heating systems.

The rate of flow of the condensate by the test strip of the new apparatus will have, we believe, a marked effect on the presence of the oxide coating and consquently on the readings of the apparatus. In other words, the removal of oxides by the flow of condensate may cause false readings and lead to questionable reasoning unless this fact is taken into consideration.

There is another item which we believe should be brought to the attention of those engaged in this general research problem. While the importance of carbon dioxide, oxygen and electrolysis should not be undervalued, we are of the opinion that the many investigators of the problems of "corrosion" in steam heating systems have overlooked the purely mechanical action of cavitation. Condensate in the bottom of steam lines and in return lines is particularly prone to this action, due to the closeness of the water temperature to the temperature corresponding to the vapor pressure. The lessening of the pressure due to change in direction of flow at a fitting or at an obstruction of any kind will cause a portion of the water to flash, form a bubble and later, on restoration of pressure, collapse with great detrimental effect to adjacent surfaces. We are of the opinion that the wasting away of many return lines as well as steam lines is entirely due to cavitation.

T. P. FINNEGAN [9] (WRITTEN): The first thing that comes to the attention of one who reads Professor Seeber's paper is that he is studying heating system corrosion by the use of an actual heating system. The results obtained by this procedure will be more convincing than the results of experiments performed in the usual laboratory

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[9] New York Steam Co., New York, N. Y.

apparatus, especially to those who are practical operating men rather than research workers. At the same time, the operation of a model system under closely controlled conditions is a difficult task and Professor Seeber and his associates are to be commended for their courage in approaching the problem from this angle.

Before any misunderstanding arises, however, it must be stated emphatically that there is nothing wrong with the work performed in the laboratory by so manv workers during the past and present. Corrosion is fundamentally a chemical problem which depends upon certain reactive properties of iron and water influenced by certain conditional factors as dissolved oxygen, dissolved carbon dioxide, and temperature. Under the same conditions, results will be found in the laboratory which will agree with the results in the model heating system. If the results do not check, it is because some variable, either unknown or uncontrolled, is operative in one case or the other.

The value of the model heating system is that the corrosive medium will not be produced synthetically by the addition of dissolved gases to water, but instead will be produced by the influence of the environment upon the condensation of the steam. As the control of corrosion in heating systems is not the attempt to remove dissolved oxygen and dissolved carbon dioxide, but rather to prevent their initial presence in the condensate, there is a valid argument for the production of an experimental condensate, not by adding dissolved gases to water but rather by controlling the steam composition and the operation of the system in such a manner as to learn the conditions of operation under which a corrosive condensate would be obtained. Such a technique, if successful, would tell not only how corrosive a condensate of a given composition might be, but also under what conditions such a condensate would be produced.

The conditions which are effective in influencing corrosion are quite well known from laboratory studies. The environments which produce these conditions might be learned from a model heating system.

It is known that the composition of condensate produced in a heating system depends to a great extent upon the design of that system. This means that Professor Seeber's results will be specifically applicable only to his own system. It is quite probable, however, that, in a general way, they may permit of interpretation with regard to all systems. For instance, they may serve to establish, to the satisfaction of all concerned, a maximum content of dissolved oxygen and dissolved carbon dioxide in the steam furnished by district steam producers.

The next logical step in this discussion is to inspect the data of the present report to learn to what extent they can be used for the purpose which has been described.

First of all it was the experience of the discusser that he could not plot these data to obtain any simple relation between dissolved gas content and corrosion rate. This would indicate that other variables are influential, and in their present state the data do not permit the evaluation of these variables. Secondly, it became evident that control of the experiments is a difficult task. It appears that in these experiments one had to take the experimental conditions as they came without being able to do very much to influence them. The result of such an inspection is that a fine start has been made on a difficult problem and that enough work has been done to point out the way for future procedures which will be of probable practical importance.

The first procedure that should be followed is to devise means of operating the system so that steam of a desired composition could be prepared throughout a complete test period. It should be possible to vary the steam composition for use in subsequent test periods.

Another procedure is to devise a means of operating the system so that the effect of variables other than those under immediate study can be controlled by making

them constant. For instance, when the effect of dissolved gases is being studied temperature should not be allowed to vary at will.

When these preliminary procedures have established the operating technique, then the collection of practical test data can be made. By this time the operators of the experiment will have worked out satisfactory routines for collecting these data.

In their final form the data will permit of relating dissolved oxygen and dissolved carbon dioxide to corrosion rate at different temperatures and with different conditions of mechanical environment such as rate of flow. Different steam compositions may be related to different corrosion rates through the effect upon condensate composition. Perhaps matters of design or operation may permit of evaluation with the other data. It has been the privilege of the discusser to have been informed of the work which has been done by Professor Seeber and his associates since the data of the present report were collected. When the next report in this series appears it will add considerable to the clarification of the practical phases which have been mentioned.

Professor Seeber has described an interesting electrical method of measuring the rate of corrosion. This method permits of measuring corrosion in a short time and as such shall prove very useful. He is to be complimented upon introducing this device to the present work.

It must be emphasized, however, that no matter what means are used to obtain the results, the results themselves should be capable of interpretation with relation to the conditions which exist in the heating system in the manner which has been described. These are the results which the membership of this organization eagerly await.

It is the opinion of this discusser that the study incidental to the development of the electrical testing method might make a good paper by itself. It is desirable that the work with the model heating system be not set aside in order to work upon the development of corrosion testing methods unless Professor Seeber decides that the customary means of determining corrosion rate are not satisfactory for his research.

A similar observation might be made regarding the correlation of the determined corrosion rate with life of piping. As both the electrical resistance method and the *N.D.H.A.* method test the water primarily, and only indirectly test the metal, there are so many difficulties in the way of arriving at a successful conclusion of such correlation experiments that this observer feels constrained to advise him to set aside this work for the present and to confine his major attention to the tests with the heating system.

The foregoing discussion is offered as a frank criticism which will be accepted in the spirit in which it is offered. A scientist has been defined as one who is searching for the truth. We are all trying to live up to this ideal. This discusser is familiar enough with Professor Seeber's accomplishments in other work to feel confident that he will help us to find answers to some of the vague questions which now confront us.

W. H. DRISCOLL: By its very nature, a research paper of this kind cannot be made quite as fascinating as a novel and, in its presentation, it is rather difficult to hold the interest of an audience. This is no reflection on the author because the statement is true of practically every research paper. The statistics and formulae necessarily included are not easily absorbed nor assimilated from the passing glance at the charts that one obtains in such a meeting as this. For this reason the value of this fine paper is likely to be lost sight of and, in order to bring more graphically to your attention a better appreciation of the importance of this problem of corrosion, I have brought with me a miscellaneous assortment of samples mounted on a panel and labeled for identification. They are here for your examination and as a basis for my discussion.

The author of this paper does not pretend to have found a remedy for the problem of corrosion but presents a report of a highly important research activity, intelligently directed, and I am hopeful that it will be continued as long as may be found necessary to determine the underlying causes of corrosion under different conditions of service, and the remedies to be applied. Frankly, the problem has me floored and, in making this statement, I am referring to real problems and not imaginary ones.

The samples I have here happen to have come from the Waldorf Astoria Hotel, in New York, because these samples happened to be available to me at this time, but I might have presented samples from any one of a number of buildings in which the management has this problem to contend with.

The widespread interest in the problem, in New York City at least, was very forcibly brought to my attention a few years ago when the New York Chapter, collaborating with the *A.S.M.E.,* the New York Real Estate Board, and the *Building Owners and Managers Association,* held a symposium on the causes of corrosion in heating systems. I was invited to preside at the meeting and, being under the impression that interest in the subject was limited and that the audience therefore would be small, I arrived at the meeting room only a few minutes before the meeting was to be called to order. To my astonishment, there was hardly space left for me in the room and I had some difficulty in reaching my seat. It was the largest Chapter meeting I have ever seen with an audience of something over 300, in a room that could not comfortably seat over 250. There were many distinguished men in the audience; physicists, research engineers, consulting engineers, professors of engineering, as well as operators and managers of some of the largest buildings in the city. There was only one paper presented, but the interest in the subject was so great that the discussion continued for almost four hours. There was a very definite opinion held by many of those in the audience, particularly the operators of plants, that the basic cause of corrosion was in the compounds or chemicals used in the boilers of the District Steam Company. They pointed to the fact that corrosion difficulties in the city occurred almost without exception in buildings supplied with District Steam, whereas buildings operated for many years with private plants were having no such difficulty. I do not agree with this theory, although it is difficult to combat it. There is undoubtedly something in the fact that, in using district steam, the condensate is all discharged to the sewer, so that the steam is constantly being generated from raw water.

New York City water is relatively pure, in the sense that it has a low content of dissolved mineral salts and organic matter and its corrosive effect would be infinitesimal in a private plant where the condensate is being constantly returned.

In a district steam plant, however, the accumulative effect of the continued use of raw water might be sufficiently great to cause some of the troubles.

From such experience as I have had, however, it appears that no serious corrosion takes place in the heating system of the building nor, for that matter, in the high pressure systems, but is confined largely to the returns from intermediate pressure systems, such as kitchens, laundries, hot water heaters, etc.

You will probably recall that one of the charts shown by Professor Seeber indicated that as the temperature increased to a certain point the corrosion rate increased correspondingly. Beyond that point, it fell off rapidly. It may be that certain critical temperatures affect the corrosion rate, or it may be, as Mr. Speller has suggested to me, that in returns from intermediate pressure systems, such as have been mentioned, the runs are shorter, the pipes smaller, and the condensate load many times greater, than would be encountered in ordinary heating systems. It is very obvious that these are conditions that generally obtain in intermediate pressure systems. Whether or not they constitute the cause of excessive corrosion has not as yet been definitely determined.

Whether or not excessive corrosion in intermediate pressure systems is due to the

fact that float traps are generally used and the carbon dioxide and other corrosive gases are pent up in the return systems and accumulate to an extent that causes the trouble, whereas, in an ordinary heating system in which a vacuum pump is generally used these gases are quickly removed with the air, are things that are entirely beyond my comprehension.

I sincerely hope that these studies will be continued and that we will have further advance reports, because I am sure that this research work will ultimately lead to an understanding and solution of this vexing problem.

F. E. GIESECKE: Corrosion of pipes is sometimes directly the result of the mineral content of the water involved. At College Station, Texas, the tap water contains about 1500 parts per million total solids, of which about 1000 parts are sodium bicarbonate. When this water is heated to about 140-160 F, the bicarbonate breaks down and liberates carbon dioxide, which is carried with the steam into the heating system and passes from there with the condensate into the return lines. While the carbon dioxide is present as a gas in the flow lines or in the radiators, it does comparatively little harm, but, after it has passed out of the radiators and is carried in solution by the condensate, it causes rapid corrosion and, in that way, the lower part of the pipe of the return lines along which the condensate flows is destroyed very rapidly. With larger pipes, it is customary to turn them through an angle, after one portion has been corroded, so as to expose a new surface to the action of the condensate containing carbon dioxide in solution.

The iron oxide which is formed is carried in solution until the condensate cools sufficiently to permit escape of the free carbon dioxide, when it deposits out as rust. In horizontal lines the rust deposits at the lower surface; in vertical lines it deposits along the entire surface. We have had cases of 1¼ in. vertical return risers which were almost completely closed; the rust had been deposited in a manner similar to the annual rings in trees. The pipe where this deposit occurred was not corroded, the rust deposited there had been brought from places nearer the radiators where corrosion was severest. In other cases, where changes of direction or changes in pipe sizes occurred, deposits were formed to such an extent that the pipes were practically closed.

W. F. SMITH: Mr. Driscoll's remarks about corrosion of pipe are very interesting and timely. At the present time we are replacing several miles of underground pipe in Philadelphia, which was used for a central hot water heating system and for domestic hot water.

In this case we found the inside of the piping for the water heating system in very fine condition, but the outside has become rusted and pitted from water outside the pipe lines working its way through the covering and attacking the exterior of the pipe. The domestic water lines were pitted on the inside. We have decided to replace the heating lines with ordinary steel pipe and replace the domestic water lines with cement lined pipe.

From Mr. Driscoll's results secured in the New York hotel, it is questionable whether the cement lined pipe is the correct material to use.

It is, of course, the end of the piece of pipe where it is threaded which always exposes the metal, and it should be doped to prevent corrosion. This is a human element, and if a spot is missed then the cement lined pipe is no better than ordinary galvanized pipe.

The installation which we are replacing has been in use for about thirty years so that there is very little complaint on the service of the same.

L. P. HYNES: It seems to me studies of corrosion where heat and moisture are present should also take into consideration electro-chemical actions. Wherever dissimilar metals are used we are likely to find one of them being seriously attacked. This has been experienced, particularly when using aluminum parts inside of Everdur

or copper hot water boilers, in which case the aluminum is very soon destroyed although with galvanized steel boilers the aluminum stands up very well.

Even where only one material is used such as steel, electro-chemical action sometimes takes place between various parts, even various sections of the same piece, and particularly where stress occurs such as the tension on the threads of a piece of pipe. I believe that puzzling cases of corrosion may sometimes be due to electro-chemical action, particularly where high temperatures are encountered.

N. D. ADAMS: At the present time the Franklin Heating Station, located in Rochester, Minn., supplies service to a group of buildings which were previously served by individual heating plants. Before connecting to the central system, these buildings experienced considerable trouble with corrosion in the return lines from the medium pressure equipment such as cooking utensils and sterilizers. Observations indicated noticeable corrosion in the return lines, particularly in those horizontal connections between the radiators and vertical risers in the return system. In those systems where a good vacuum was maintained, there was less corrosion in the returns, which would indicate that the removal of air assisted in the elimination of corrosion. In those cases where the returns from the medium pressure lines were connected to the vacuum heating system receiving tank through a trap and condenser, so that the air in the medium pressure returns was evacuated, less trouble was experienced in the medium pressure lines.

Brass pipe was formerly used between the boiler and the flue gas analyzer, and we formerly experienced considerable difficulty in maintenance due to its dezincification by the carbon dioxide. Later the brass pipe was replaced with a pure copper pipe and no further trouble has been experienced.

The water used in the domestic hot water system was obtained from deep well water containing 16 to 17 grains of sodium bicarbonate and softened by the Zeolite process. The quantities of water used in this system were measured by meters containing aluminum discs. These discs would last about 6 months due to the action of the sodium. Finally this trouble was eliminated by replacing the aluminum discs with composition discs reinforced with metal.

The feed water heater used in this system is of the deaerating type. The returns from the heating system are constantly tested in an effort to maintain a pH reading of 7.4 or higher on the alkaline side. This is accomplished by making a daily chemical analysis of the boiler water and introducing such chemicals as are deemed necessary. Under these conditions, little active corrosion has been observed.

I wish to compliment Professor Seeber and his associates on the work that has been done and the work that will be continued, as the cost of materials and labor in replacement due to corrosion is a tremendous waste which can be prevented.

No. 1038

PERFORMANCE OF AN UNDERFEED DOMESTIC STOKER

By T. G. Estep * and D. C. Saylor ** (*NON-MEMBERS*), Pittsburgh, Pa.

THE particular stoker which is the subject of this discussion consists of a coal hopper from which fuel is delivered to a continuously driven screw. The screw forces the coal through the feed tube horizontally for approximately 3 ft. The pushing action of the feed screw then elevates the coal vertically into the retort. Surrounding the upper part of the retort is an air space with openings in the retort side through which air is supplied by a fan to the fuel bed for combustion.

A small amount of air is also supplied to the end of the feed tube to prevent furnace gases from blowing back through the coal hopper. Means are provided to prevent arching of the coal over the feed screw and the hopper is also fitted with a gas tight cover.

The air supply to the fuel bed is manually controlled by a louver damper in the fan inlet. A floating butterfly damper with a graduated index is installed in the fan discharge so that the proper supply of air can be approximately determined.

The stoker unit was mounted under a 1200 sq ft capacity rectangular house heating boiler. The boiler was prepared for test work exactly in accordance with the recommendations of the American Society of Heating and Ventilating Engineers.

The water and coal supplied the unit were determined by weighing. The tests were started with the furnace hot and conditions constant. Carbon dioxide readings were recorded on a gas analyzer, and the flue gas composition checked with an Orsat apparatus. Steam pressure was maintained constant with an automatic back pressure valve.

The boiler used had a grate area of 8.2 sq ft when arranged for hand firing. When the stoker was installed, the area at the top of the retort was 0.492 sq ft. The top retort area was used in determining the pounds of coal per square foot of grate area since the air was distributed to the fuel bed mainly through this area. This neglected the slow burning of the coke thrown clear of the retort top. The furnace volume after the stoker was installed equalled 14.84 cu ft.

Two types of coal, a high and low volatile bituminous, were used in securing

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the results which are shown graphically in Figs. 1 and 2. The coal size was ⅜ x 1⅛, the size recommended by the stoker manufacturer. These coals had the analyses as given:

PROXIMATE ANALYSIS, AS RECEIVED

	HIGH VOLATILE	LOW VOLATILE
Free moisture	1.8	2.0
Volatile matter	35.0	16.7
Fixed Carbon	52.8	74.0
Ash	10.4	7.3
	100.0	100.0

ULTIMATE ANALYSIS, AS RECEIVED

	HIGH VOLATILE	LOW VOLATILE
Hydrogen	5.10	4.51
Carbon	73.80	80.00
Nitrogen	1.40	1.76
Oxygen	7.40	4.73
Sulphur	1.90	1.50
Ash	10.40	7.50
	100.00	100.00
Heating value	13,260	14,500
Ash softening temperature	2,490 F	2,600 F

After several preliminary tests in which various furnace arrangements were used, it was decided to perform 3 series of tests with the furnace arrangement recommended by the stoker manufacturer. The tests were:

1. High volatile coal, no attention to the fire.
2. High volatile coal, attention to the fire.
3. Low volatile coal, no attention to the fire.

The results of the tests with attention to the fire are not included in the graphical representation of the results. The values of the various items that were obtained when the fuel bed received attention are given in the discussion, and will indicate the beneficial effects of fuel bed agitation in the performance of the stoker. The results obtained by breaking up the coke plug and raking the unburned carbon into the combustion zone are similar to the results which were obtained when working with a stoker that had positive fuel bed agitation.

When a domestic stoker is installed in a residence heating plant, its operation is controlled by various devices which maintain a reasonably constant temperature in the dwelling by intermittent operation of the stoker. To determine the performance of the stoker under these conditions would require that the test be made in some residence and the duration would be for the entire heating season. Thus only one test could be made per year. By operating the stoker continuously at a constant rate it is possible to perform a large number of tests in a much shorter time and still determine the general operating characteristics, particularly the limitations. There is no doubt that this accelerated method of testing will magnify the faults of the stoker, but this may also be desirable.

The tests herein reported were made with the stoker operating continuously with a constant rate of feed during the test. The stoker is provided with 3 rates of feed and several tests were made at each rate so as to obtain an average performance.

Since there is no ash pit in a stoker installation of this type and since coke

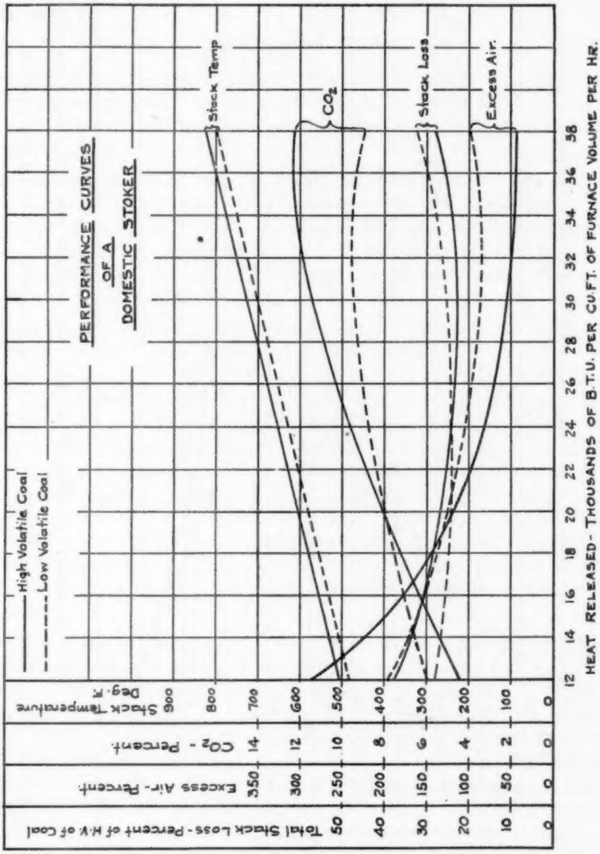

FIG. 1. STACK TEMPERATURE, CARBON DIOXIDE, STACK LOSS AND EXCESS AIR PERFORMANCE CURVES

was formed which could not be burned, there is a question as to the ash pit loss. In the early tests an effort was made to remove the coke and ashes, to separate the ashes from the coke and to determine the amount of carbon in the ash. This was not a satisfactory procedure as the results of such determinations were not consistent. The method finally used was a heat balance method. The stack gas analysis, weight of water used, the coal analysis, and the stack temperature allowed the weight of coal actually burned to be computed. This

computed weight of coal was from 1 to 8 lb of coal per hour less than the weight of coal actually supplied. Attention should be called to the fact that this represents a greater weight of carbon, since the volatile constituents of the coal actually supplied the furnace were burned.

The performance of the stoker is represented by a series of curves shown in

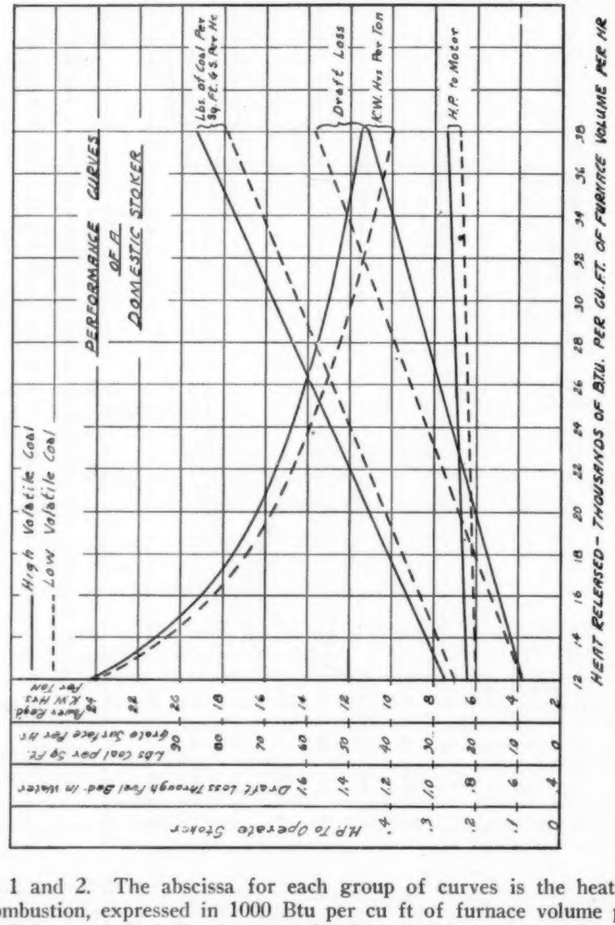

FIG. 2. COAL RATE, DRAFT LOSS AND POWER CONSUMPTION PERFORMANCE CURVES

Figs. 1 and 2. The abscissa for each group of curves is the heat released by combustion, expressed in 1000 Btu per cu ft of furnace volume per hour. The efficiency of the boiler is purposely eliminated from the results, as it was desired to obtain only the stoker performance and not the combined performance of the stoker and the boiler. Perhaps this somewhat limits the value of these tests to the heating and ventilating profession who generally judge the performance of the entire heating plant in terms of equivalent direct radiation

and this unit of measurement does involve boiler efficiency. Since the efficiency is not constant over the range of operation shown, it is not possible to use a second scale of abscissa for the same curves showing equivalent direct radiation corresponding to a given heat release. However, if it is desired to estimate the equivalent direct radiation, if the given heat release is multiplied by an assumed boiler efficiency and the product divided by 240, the desired result will be secured.

Discussion of Results

The stoker manufacturer furnished a pair of tongs to remove the ash, the assumption being made that the ash would be fused in the form of clinker. The only clinker produced was a negligible amount at the top of the retort on the hopper side.

Both of the coals used had ash fusion temperatures that are within the range of the domestic fuel coals. Also the tests were run continuously which meant that hotter furnace conditions were obtained than on intermittent operation. If the ash does not clinker under the conditions of operation, then there is a dirty disagreeable task of ash removal which will upset the conditions of operation for a considerable time.

Where the coal is elevated vertically to the level of the fuel bed, the coal passage casting is formed with a long radius bend. The curve allowed the coal to pack on this side of the casting and this action extended up to the combustion zone. Air flowed through the fuel bed with less resistance on the side away from the curve. As a result of this difference in density of the fuel bed, the rate of combustion on the hopper side of the retort was much higher than on the opposite side, and the clinker which was formed occurred in this hot zone. The small amount of clinker formed obstructed the air openings and tended to correct the uneven distribution of air. It was thought that by plugging some of the air openings on the hopper side the clinker formation might be avoided and that the air distribution would be more uniform from the beginning, but trials with some of these openings closed did not seem to improve combustion.

The screw for delivering the coal from the hopper to the retort operates continuously and at a constant speed for a given rate of feeding. Thus there is no shock or agitation to the fuel bed as there would be if the screw had an intermittent motion such as would be obtained if driven by a ratchet arrangement. As the coal passes through the retort and is partially burned, it produces a cylindrical coke plug, the diameter of which is slightly less than the diameter of the retort. This coke plug rises at an even rate of speed until it is approximately 8 in. above the top of the retort. The plug then breaks due to strains produced by cooling and some of the coke is thrown clear of the combustion zone. The coke thrown out of the fire zone does not burn readily, and, unless the fuel bed receives some hand manipulation, there are two distinctly different rates of combustion occurring in the furnace, one in the fuel remaining in the retort and the other in the fuel which has fallen aside.

The conditions mentioned previously influence the design of the fan materially for to attempt to force air through the plug will require a high head, while to supply air to burn the coke at the top of the air ring requires only a moderate pressure. The quantity of air supplied for a given rate of fuel

feed should be constant but, since the fuel bed resistance is variable, a variable amount of air will be delivered to the burning coal.

If reference is made to the excess air curves, Fig. 1, it is seen that at low ratings there is a high percentage of excess air. The percentage of excess air decreases as the rate of burning coal increases and tends to become constant at the high values of heat release. This is due to the uneven packing action of the screw feed and is especially noticeable when there is a light fuel bed. As the fuel bed becomes heavier with the increased rate of coal feeding, there is a better air distribution. If the fan forces more air through the packed heavy fuel bed than when the fuel bed is less dense, too much air will be supplied. This action will sometimes cool the coke plug and an accumulative effect results.

When the fuel bed was manipulated by hand, the excess air was materially reduced as was shown by the higher CO_2 content of the products of combustion of about 12 per cent, which indicates the effects of a more uniform fuel bed.

The CO_2 content of the flue gases is often cited as a criterion of the completeness of the combustion process. Since there was no CO present, it is apparent that the coal consumed was burned completely, but it does not indicate anything about the coke thrown clear of the combustion zone. The CO_2 curves for the two coals show that at low rates of combustion the low volatile coal gave a much higher CO_2 content than the high volatile. The low volatile coal was a non-caking variety and gave a fairly uniform fuel bed at all times. The high volatile coal was a caking type and at low rates of combustion most of the air was blown through the fissures formed in the fuel bed and was not used in the combustion process. At the higher rates of combustion the CO_2 content from the high volatile coal was more than the low volatile for two reasons; first, the volatile matter rising above the fuel bed utilized some of the excess air in the furnace atmosphere; and, second, the low volatile coal, being a slow burning coal, required a large amount of air pressure to maintain the proper combustion rate. It would seem that for burning high volatile caking coals the fan should produce a relatively high head with a corresponding small volume, but for low volatile, non-caking coals the volume should be large with a small head.

The stack temperature curves are essentially straight lines and the values do not exceed those of good practice. The curves are the results of two actions: namely, the heat released in the boiler and the heat absorbed by the boiler heating surfaces from the hot gases. The higher stack temperatures obtained with the high volatile coal are due to the longer flame.

The high volatile coal gave generally a lower stock loss than the low volatile coal. The loss is a product of the temperature difference and the weight of the gaseous products of combustion. While the stack temperature was higher with the high volatile coal, the weight of the products of combustion was enough lower to reduce the total loss below that of the low volatile coal.

The stack losses were lowered about 5 per cent when the fire was hand manipulated. This was due to the fact that manipulating the fuel bed resulted in a glowing mass of fuel. This glowing mass radiated more heat to the furnace than the top of the coke plug, and also reduced the excess air. The stack loss curves include the loss due to moisture in the coal and also the moisture formed from the combustion of the hydrogen in the fuel bed.

The power supplied the unit was almost constant, increasing slightly at the increased rates of coal feed. Most of the power supplied the unit was used in overcoming friction. For this reason the power per ton of coal was greater at the low rates of coal feed. The draft loss through the fuel bed was greater for the low volatile than for the high volatile coal. This may be accounted for by the fact that the low volatile coal, being very friable, broke up into a greater proportion of fines which gave a more compact fuel bed. The high volatile coal, because of its caking properties, produced large fissures in the fuel bed which offered little resistance to the flow of air.

CONCLUSIONS

1. The stoker is satisfactory in its operation provided a non-caking variety of coal is used. The coal should have an ash fusion temperature low enough to fuse the ash so it can be readily removed.

2. Caking coals will be hard to manipulate with this stoker.

3. From the results of these tests and other tests, it is evident that the action of the stoker may be improved by the proper design of fan.

4. The long radius bend in the coal feed passage is responsible for faulty air distribution in the fuel bed.

5. If air could be introduced into the center of the fuel bed, the probability of the coke plug forming would be lessened.

6. The best results were obtained when hand attention was given to the fuel bed.

DISCUSSION

R. A. SHERMAN (WRITTEN): The authors of this paper are to be commended for their judgment in separation of the functions of the stoker and of the boiler in which it is installed and their omission of values for overall thermal efficiency in the report. The function of the stoker is to liberate the heat from the coal, and that of the boiler is to absorb the heat liberated. An overall efficiency value often leads the uninformed to credit the stoker or the fuel with a high efficiency or to blame them for a low efficiency which is properly due to the boiler.

The authors have wisely titled their paper "Performance of *an* Underfeed Domestic Stoker," and the writer wishes to emphasize the fact that the results apply only for the particular installation and method of operation, for both of these were such that the results cannot be considered as typical of the expected performance of an underfeed stoker using bituminous coal.

The first and most important reason why the results cannot be considered as applicable to underfeed stokers as used for domestic heating is that the burning tests were conducted with continuous rather than intermittent operation. No general prediction can be made as to whether the results under continuous operation will be better or worse than with intermittent operation, but, inasmuch as all domestic stokers do operate intermittently, the results are no index of expected operating results in the home. A test over an entire heating season such as the authors mention is by no means necessary. The least that could have been done would have been to control the *on* and *off* operation manually according to a definite time schedule, and a better and very simple means would have been to control the operation by a switch turned off and on between certain limits of boiler pressure. This would have closely approximated operating conditions.

The authors call attention to the segregation of sizes and non-uniform feeding of

the coal to different parts of the retort. A previously published report[1] gave considerable data on the segregation of sizes and uniformity of feeding and pointed out that segregation was quite severe with the size, 1⅛ x ⅜ in., coal which the authors used. They have pointed out that better combustion could be obtained with manual leveling of the fire. Any stoker that does not operate satisfactorily without manual attention other than charging of the coal to the hopper and the removal of the clinker cannot be said to be satisfactory.

If a stoker does not meet these requirements, it may be because of improper installation; the installation used by the authors could probably have been improved. No drawings are given, but the description indicates that the retort of only about one-half square foot in area was installed in the middle of a flat hearth about 8 sq ft in area. Naturally, the coke that broke off from the column that was pushed up into the center of the retort could fall onto the hearth out of the range of the air admitted through the tuyeres. In the Fuel Research Laboratories of Bituminous Coal Research at Battelle Memorial Institute, we have found that, with a round combustion chamber with walls that closely restrict the fall of the coke, the coke will fall back into the retort without manual attention and produce excellent combustion results even with coking coals.

If the tests had been for a prolonged period, the coke and ash pushed to the sides would have tended to form, even in the rectangular hearth, a dish-shaped contour that would have produced a similar result to the round combustion chamber with walls close to the retort. This condition can be simulated and good results produced in a shorter time if the hearth is made dish-shaped when installed. Data are lacking in the paper as to the length of the burning tests. If the time of each test was relatively short, as it possibly was, the reason for the accumulation of coke outside the retort is obvious.

If the tests were of short duration, this would also partly account for the failure to produce satisfactory clinker. The ash-softening temperatures reported, although relatively high, are within the range in which satisfactory operation is being obtained on this type of stoker. It will not be obtained, however, in a period of 8, 12 or 24 hours. In a hearth of the size used there is ample room outside the retort for the accumulation of the ash from a considerable amount of coal. This will probably accumulate in the first day or so of operation as loose ash, but when it has formed the dish-shaped contour the ash liberated later will remain near the periphery of the retort where it will be subjected to the intense heat of combustion and will fuse into a clinker ring that can be readily removed with the tongs. As these stokers are designed for the removal of the ash as clinker, a type of operation that does not produce a hard clinker cannot be considered satisfactory.

The authors report that the unburned carbon in the loose ash that they removed was high, which is expected with the type of operation that they obtained. The writer has found from field tests in homes and from laboratory tests that the unburned carbon in the ash removed as clinker is practically negligible.

The paper reports a wide range of excess air with change in rate of operation of the stoker. Just why this is necessarily so is not evident to the writer. The rate of coal feed and the rate of air supply on this type of stoker are independently variable. Within very wide limits, one can adjust the ratio of air to coal as one wills and thus obtain variable amounts of excess air at any rate of operation of the stoker. The authors state that the fuel bed became thicker at the higher rates of coal feed. This was not necessarily so but merely indicates that they reduced the ratio of air to coal as they increased the rate of coal feed. It is apparently not generally recognized that the depth of the fuel bed is neither an independent variable nor fixed solely by the rate of coal feed. With a given coal and a given stoker and

[1] Technical Report No. 1 of Bituminous Coal Research, Inc., December, 1935.

furnace, the depth of the fuel bed is a function of the ratio of the air supply to the coal supply. Having fixed this ratio, the depth of the fuel bed will be fixed.

The paper mentions that as the fuel bed tended to become thicker the resistance built up and thus less air was supplied and an accumulative cycle resulted. Obviously, the stoker was not equipped with an automatic air volume control. Many modern stokers are now equipped with such controls that are effective in maintaining over a wide range of fuel bed resistance a constant supply of air from the fan. This automatically corrects the accumulative effect and maintains very constant depth of fuel bed and excess air.

Due, therefore, to an undesirable method of installation, to the continuous rather than intermittent operation, to possibly a too short period of tests, and to the inadequate interpretation of some of the results, the data presented in this paper and the conclusions at its close must be considered only as of interest and value for the particular conditions used. They cannot be considered as indicative of the performance of the typical domestic underfeed stoker which is able to burn satisfactorily bituminous coals which have a wide range of caking and coking characteristics.

D. W. NELSON (WRITTEN): The tests reported in this paper recall some tests made at the University of Wisconsin on a 3200 sq ft capacity steel heating boiler. A summary of some of these tests is given in Table A. Results are shown for one test with hand firing, three tests with an underfeed stoker, and one test with an oil burner. Some 30 tests were made at various ratings and about 12 tons of coal were burned in all of them. All tests were made under continuous firing conditions.

The stoker was of the intermittent feed type rather than the continuous type mentioned in this paper. It was installed with a clearance height of about 26 in. and a furnace volume of about 30 cu ft. In preliminary tests a coal was tried that caked

TABLE A. SUMMARY OF RESULTS OF TESTS ON STEEL HEATING BOILER

No.	ITEM	FUEL BURNING DEVICE				
1	Type of Firing	Hand	Stoker	Stoker	Stoker	Burner
2	Rating of Test	110.0	110.2	106.3	102.5	105.2
3	Kind of Fuel	A	B	C	D	F
4	Size of Fuel	6 × 3	Nut	Nut	Nut	(Oil)
	Proximate Analysis					
5	Moisture, per cent	6.95	7.36	7.54	11.82	....
6	Volatile Matter, per cent	34.35	30.05	33.00	35.80	....
7	Fixed Carbon, per cent	53.30	55.01	53.77	46.71	....
8	Ash, per cent	5.40	7.58	5.07	5.67	....
	Ultimate Analysis					
9	Hydrogen, per cent	8.81	8.98	8.94	8.42	12.85
10	Carbon, per cent	78.65	76.88	78.60	77.67	84.27
11	Nitrogen, per cent	1.53	1.54	1.55	1.48	2.52
12	Oxygen, per cent	11.01	12.60	10.91	12.43	....
13	Sulphur, per cent	....	....	....	....	0.36
14	Ash, per cent	5.81	8.19	5.67	6.43	....
15	Heat Value as Fired, per pound	12,583	12,400	12,800	11,820	19,712
16	Draft Loss in Fuel Bed	0.003	0.446	0.563	0.909	....
17	Carbon Dioxide, per cent	13.5	12.7	12.9	10.3	11.2
18	Oxygen, per cent	4.0	4.7	4.9	5.7	4.3
19	Carbon Monoxide, per cent	0.0	0.0	0.0	0.1	0.0
20	Nitrogen, per cent	82.5	82.6	82.2	83.9	84.5
21	Temp. of Flue Gas	676.0	662.0	557.0	540.0	492.0
22	Fuel per Sq Ft Grate per hour, per pound	11.91	15.02	13.76	15.17	....
23	Output, Btu per hour	935,000	955,000	917,000	879,000	896,000
24	Equiv. Evap. per pound	7.35	8.95	9.81	8.15	14.53
	Heat Balance, per cent					
25	Absorbed by Water (Overall Efficiency)	56.7	70.0	74.3	66.9	71.5
26	Heat Lost in Moisture	8.6	8.9	8.5	8.6	7.5
27	Heat in Dry Flue Gases	16.58	17.42	16.88	21.35	9.5
28	Heat Lost in CO	....	....	....	....	....
29	Carbon Loss in Ash	0.15	0.04	0.10	0.20	0.0
30	Radiation and Unaccounted for (by Difference)	17.97	3.64	0.22	2.95	11.50

badly in the burning process or rather in the liberating of the volatile matter so that it was impossible to carry even normal rating of the boiler due to interference with the air distribution. Since only continual manual breaking up of the fuel bed made it possible to carry a reasonable load, this coal was considered not suited to this stoker installation. The slight agitation due to intermittent feed of the screw seemed to have no value towards breaking up of the caked bed. The three fuels in the three stoker tests shown in the table gave good results. The loss due to carbon in the ash was negligibly small in all tests. Clinkering was satisfactory except in some of the light load tests. The power used per ton of coal in the three stoker tests shown was 7.25 kwhr for B, 6.23 kwhr for C, and 7.42 kwhr for D. On preliminary tests it was difficult to obtain satisfactory CO_2 values due to the coal feeding towards the rear of the retort. The placing of a low brick wall at the rear of the retort aided distribution and prevented excess air entering at the front of the retort.

The tests with the stoker showed an overall efficiency of 67 to 74 per cent at from 100 to 120 per cent of normal rating for the boiler. At higher outputs the efficiency dropped due to reduced efficiency of heat absorption as indicated by higher flue gas temperatures. The boiler reached its maximum capacity as limited by carry over water with the steam at 150 per cent of rating. At lower ratings the efficiency dropped to between 57 and 65 per cent due to poorer combustion conditions as indicated by lower CO_2. The maximum output of 150 per cent of rating was also secured with hand firing with attention averaging every half hour and firing every 1.2 hr. On these hand fired tests the tubes had to be cleaned at times in order to continue to carry the load. On the stoker tests the tubes were cleaned only between tests.

Under actual heating conditions the efficiency at partial loads would likely be lower than secured in these tests due to intermittent operation. A considerable part of the fuel is burned during off periods in small installations. It is a difficult problem to adjust the air supply for various fuel bed resistances and for various lengths of off-period burning. The stoker industry is paying an increased amount of attention to refinements in the control of the air supply. The smallest size of stoker made by many manufacturers is too large for the average residence. This results in very short running periods, with an overloading of the combustion space and heat absorption surface and long off periods. As with oil burners it would seem that the maximum capacity of domestic stokers should not greatly exceed the maximum demand.

The paper under discussion has considerable value in leading towards a more technical approach in the domestic stoker field.

W. H. SEVERNS: Were any attempts made to measure the amounts of air which passed through damper openings of the firing door to enter the furnace above the fire? Were measurements made of the quantities of air, supplied by the fan, which passed through the tuyeres into the fuel bed?

L. P. HYNES: The paper is interesting and I believe the comments about the need for introducing an ample air supply to the interior of the fuel bed are well borne out by European experience. Some very successful results have been obtained in England, France, and other countries with a French design of stoker whereby the air supply is carefully controlled and is fed by jets to the interior of the fuel bed, resulting in an improved combustion over similar stokers without this air control.

No. 1039

RELATIVE ABSORPTION OF THE VARIOUS SECTIONS OF A ROUND HEATING BOILER

By A. J. Johnson * (*MEMBER*) and P. A. Mulcey ** (*NON-MEMBER*)
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An understanding of the relative value and exact part played by each individual section of a vertical or round heating boiler is of considerable importance for a number of reasons. A knowledge of the increased efficiency and output which may be expected as the result of adding one or more intermediate boiler sections is of definite value both from an economic and engineering viewpoint. The proportion of radiant heat absorbed by the furnace side walls and by the crown sheet is useful in determining the proper fuel bed depths to be carried with various solid fuels. The effect of forcing flame to impinge upon boiler side walls gives definite information concerning the practical value of furnace baffles.

A study of heat absorption from fuels, such as coal and coke, which have compact incandescent fuel beds, as compared with liquid or gaseous fuels with which heat is generated, without the use of a fixed fuel bed, is also of interest.

Many other aspects of an investigation of this type also prove useful in designing and proportioning heaters. For this reason every effort was made to secure results which would be typical of actual conditions such as might be repeatedly encountered in the field.

Description of Apparatus

The tests were conducted in a 25 in. five-section cast iron boiler. The firepot section, approximately 2 ft in diameter, was tapered slightly inward at the top with an overall depth of 24 in. The flue passes within the intermediate sections were approximately 3 in. deep so that the initial gas velocity was several times greater than in the furnace. The absorption surfaces of the several sections were:

Firepot Section	13.5 sq ft
Crown Sheet	11.3
Each Intermediate (2 Sections, 11.0 sq ft each)	22.0
Dome Section	7.0
Total	53.8 sq ft

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Presented at the Semi-Annual Meeting of the American Society of Heating and Ventilating Engineers, June, 1936, Buck Hill Falls, Pa., by A. J. Johnson.

Connecting nipples between each section as well as between the furnace side wall section and crown sheet were filled with lead so that water could be passed through each unit separately and its heat rise measured.

The outlet temperature from each section was graduated approximately from 120 F in the firepot section to 160 F in the top section in order to simulate

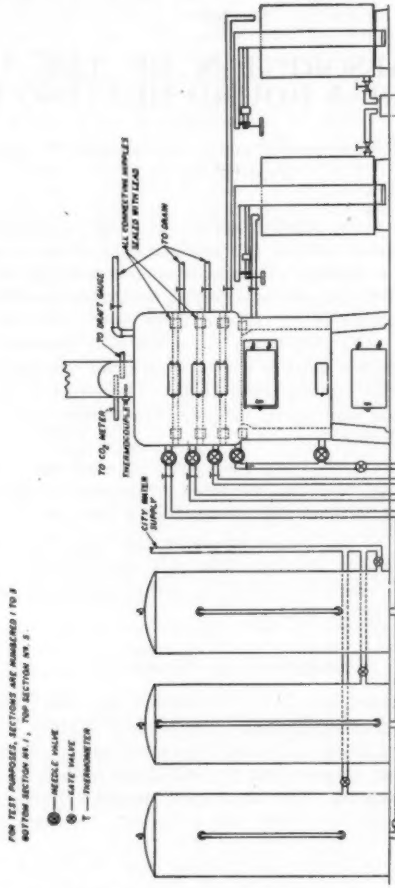

FIG. 1. DIAGRAM OF APPARATUS ARRANGEMENT FOR ABSORPTION TESTS

normal conditions in which the hottest water would be at the top of the heater. Even with this precaution, however, boiler efficiencies were somewhat higher than normal.

Various fuels were used, including anthracite, by-product coke, petroleum coke, oil, and gas. Oil was fired in a rotary type burner, and the gas was consumed in a conversion type burner.

Hand fired tests were all operated as cycle tests with all the usual readings and measurements for determining the efficiency and conditions of combustion. The automatic equipment tests were 8 hour, running start tests.

A graphic picturization of the testing layout is shown in Fig. 1, whereas the log sheets given in the discussion of this paper show the draft and fuel bed conditions of the various tests.

Average Absorption with All Fuels

As expressed in terms of percentage (with the total output of the heater as 100 per cent),[1] the absorption of each section remained surprisingly constant with each of the various fuels. Table 1 and Fig. 2 show the general average results with all fuels, whereas Table 2 gives individual results with each fuel.

Table 1. Absorption of Each Section Averaged for All Fuels

Side Wall Section	56.9%
Crown Sheet Section	17.9%
1st Intermediate	13.1%
2nd Intermediate	7.9%
Dome Section	4.2%
Total Boiler Output	100.0%

The only plausible explanation of the similarity of the distribution of heat absorption throughout the boiler when using solid and fluid fuels would seem to be that a sufficient amount of heat is radiated from the glowing mass of burning oil or gas to balance the radiant heat from a bed of solid fuel.

To substantiate this conclusion it will be noted that the results with an underfeed stoker without a baffle were of an entirely different character, in that the absorption of the side walls was lower than the average by some 17 per cent, whereas the crown sheet absorbed 11 per cent more than the all-fuel average. In this type of stoker the fuel bed is relatively flat and is constructed in such a manner that nearly all radiant heat will be transmitted to the crown sheet rather than at an oblique angle to the side walls. The flame from the stoker also differs from oil and gas in that it is probably merely the luminous products of combustion rather than the actual burning mass of an appreciable weight of fuel.

Proportion of Heat Absorbed

In the first or side wall section of the boiler, absorption by conduction, radiation, and convection can and probably does occur simultaneously. In the crown sheet, only radiation and convection are possible (unless a flame from actual burning fuel impinges upon the crown sheet). In all other heating surfaces, absorption is limited to that from convection.

In oil and gas furnaces absorption by conduction is insignificant, whereas even in burning solid fuels these tests indicated that it is a negligible quantity. The *point to point* contact of solid fuels against the side walls does not offer any large or very effective surfaces for transmission of heat by conduction.

[1] Averages in this section refer to per cent of actual total heat absorbed by boiler and are not to be confused with efficiency results which would of course refer to per cent of input.

The conductivity of fuels is relatively low, so that there is little likelihood of a free flow of conducted heat horizontally through a fuel bed, and accumulations of ash and dead fire against the side walls act as an insulator.

While it is difficult and impractical to secure actual measurements which would segregate radiated and conducted heat to the side walls, calculations

TABLE 2. RELATIVE AMOUNT OF HEAT ABSORBED BY EACH BOILER SECTION
(*Expressed as Per Cent of the Total Heater Output*)

Fuel	No. of Tests	Boiler Part					Total
		1 Side Walls	2 Crown Sheet	3 First Intermediate	4 Second Intermediate	5 Dome	
Anthracite [a]							
Egg	6	56.8	17.0	13.1	8.3	4.8	100.0
Stove	6	57.9	16.9	13.0	7.8	4.4	100.0
Chestnut	10	58.1	17.8	12.5	7.8	3.8	100.0
Pea	6	56.4	17.4	13.9	8.0	4.3	100.0
Buckwheat [b]	5	57.0	18.0	13.8	7.5	3.7	100.0
Ave. [c]	..	57.3	17.4	13.2	7.9	4.2	100.0
Anthracite Stoker [d] with Baffle	2	55.4	18.4	13.2	8.5	4.5	100.0
Anthracite Stoker [e] without Baffle	4	38.9	30.1	15.8	9.4	5.8	100.0
Petroleum Coke	6	57.8	14.9	12.9	8.7	5.7	100.0
By-Product Coke	6	55.2	16.5	14.0	9.4	4.9	100.0
Oil Burner	4	52.0	21.8	15.1	7.7	3.4	100.0
Gas Burner [f]	4	64.1	18.6	10.0	5.2	2.1	100.0
Average [g]	..	56.9	17.9	13.1	7.9	4.2	100.0

[a] Hand fired only.
[b] Induced draft.
[c] This average includes hand fired anthracites only.
[d] With 21 in. baffle 10 in. above firepot.
[e] Not included in average because of different nature.
[f] Including refractory brick.
[g] Includes all fuels except stoker without baffle.

nevertheless substantiate this conclusion, since little unaccounted absorption remains after deducting the calculated radiant and convected heat.

Assuming a uniform temperature over the surface exposed vertically to the furnace walls and to the crown sheet, the calculated radiation to the firepot or side walls would be from 3.5 to 6 times greater than that to the crown sheet. This ratio would, however, be considerably reduced if, as is undoubtedly the case, the upper part of the fuel bed is hotter than the lower. Substantiating these calculations, in all hand fired tests, the actual measured absorption of the side walls was from 2.4 to 3.6 times greater per square foot than that absorbed in the crown sheet section.

This indicates that most of the heat absorbed by the side walls is by radiation rather than by convection, since if the latter were the case the heat absorbed per square foot would probably be greater for the crown sheet because of the sweep of hot gases across its surfaces.

As compared with radiation, convection is extremely difficult to calculate because of such variables as gas velocity, etc. Therefore, in order to determine

the amount of heat absorbed in each section by convection, the total heat actually absorbed in the tests was plotted as measured. Radiation was then calculated and plotted on the same curves. Neglecting conduction, and it is probably negligible for reasons which have been explained, the difference between this actual total absorption and the absorption by radiation will represent convection. The heat absorbed by each section in typical individual tests is shown in Figs. 3 and 4. Total absorption as measured, radiation and convection as calculated, were plotted as at the midpoints of each boiler section upon abscissae representing square feet of heating surface. As the ordinates represented heat absorbed per square foot, the area of the shaded blocks is proportionate to the heat absorbed by each section. Fig. 3 is a chestnut anthracite test at a rate of 4 lb per square foot per hour, whereas Fig. 4 is the same fuel but at the higher rate of 8.6 lb per square foot per hour.

ECONOMIC VALUE OF ADDED BOILER SECTIONS

Aside from the design standpoint, one of the most important uses of this type of information is a determination of the economic value of added boiler

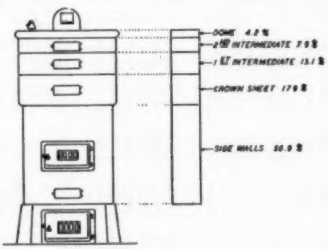

FIG. 2. ROUND HEATER WITH EACH SECTION DRAWN IN PROPORTION TO AMOUNT OF HEAT (AVERAGE) WHICH IT ABSORBED

sections. The side walls and at least a combined crown sheet and dome are of course indispensable. It is possible, however, through the addition of one or more intermediate sections, to increase the absorbing power of the boiler to a limit determined only by the economic ratio between the output to be expected and the cost of the added metal.

In the case of the present tests, the sensible heat carried off by the stack gases is in part recoverable by the addition of sufficient sections to reduce the temperature of the gases to about 212 F. This boiler, however, is already equipped with two intermediate sections having corrugated heating surfaces and the addition of further sections can be readily shown to be both uneconomical and impractical. In Table 3 the test (No. 2230-J) with chestnut coal at maximum output shows 9.5 per cent *recoverable* heat in the stack loss. Assuming absorption to be as rapid as in the top section (and this is not strictly true because of the fact that the curve of absorption flattens out as its abscissa increases in length, as shown in Figs. 3 and 4), to recover this heat would require the addition of about 28 sq ft of absorption surface, increasing the present boiler size over 50 per cent.

Furthermore, at lower outputs this recoverable loss becomes nearly negligible, being less than 2½ per cent at one-half boiler rating.

TABLE 3. RESULTS OF ALL TESTS SHOWING CONDITIONS OF FUEL BED DEPTH, DRAFT, OUTPUT AND PER CENT OUTPUT OF EACH SECTION

Test No.	Fuel Bed Depth In.	Stack Draft	Duration Hours	Coal Per Hour	Overall Eff.	Total Output Btu	Output, Btu Per Hour						Output, Per Cent of Total					Absorption, Per Cent of Total Input					
							Total	Sec. 1	Sec. 2	Sec. 3	Sec. 4	Sec. 5	Sec. 1	Sec. 2	Sec. 3	Sec. 4	Sec. 5	Sec. 1	Sec. 2	Sec. 3	Sec. 4	Sec. 5	Stack[a]
SECTION I. ANTHRACITE—EGG SIZE																							
2230-BD	16	0.03	8.580	19.36	64.8	1,462620	170220	99700	28700	21820	12600	7400	58.60	16.85	12.82	7.39	4.34	38.00	10.92	8.31	4.79	2.82	11.00
2230-BE	16	0.06	8.330	21.56	70.1	1,712654	205000	120000	34200	25550	15660	9590	58.60	16.67	12.42	7.64	4.67	41.10	11.68	8.70	5.35	3.27	12.10
2230-BF	16	0.09	8.250	22.32	71.5	1,785047	216380	121500	37750	28700	18350	10080	56.20	17.42	13.25	8.48	4.65	40.20	12.46	9.47	6.06	3.32	14.60
2230-BG	16	0.12	6.583	28.35	68.8	1,737400	264130	153000	43400	33400	21850	12480	57.90	16.44	12.64	8.28	4.74	39.80	11.30	8.70	5.69	3.26	13.90
2230-BC	12	0.06	6.550	21.05	65.2	1,217942	186010	101700	34000	25100	16700	8510	54.70	18.30	13.50	9.00	4.50	35.60	11.92	8.80	5.86	2.93	15.00
2230-BB	12	0.12	5.300	26.30	65.0	1,228080	231950	127500	38200	32350	20700	13200	55.00	16.50	13.85	8.93	5.72	35.80	10.73	9.00	5.80	3.72	16.00
SECTION II. ANTHRACITE—STOVE SIZE																							
2230-BK	16	0.03	12.000	17.60	72.9	2,098472	174750	104800	31100	21100	11440	6310	60.00	17.80	12.10	6.55	3.55	43.75	12.97	8.82	4.78	2.59	8.80
2230-BI	16	0.06	9.833	20.80	72.5	2,020307	205300	121000	35700	25300	15000	8300	59.00	17.36	12.30	7.31	4.03	42.80	12.59	8.91	5.30	2.92	10.20
2230-BM	16	0.09	7.417	26.40	66.7	1,775576	239450	140500	38480	30900	18540	11030	58.62	16.08	12.93	7.74	4.63	39.10	10.70	8.63	5.16	3.09	11.00
2230-BL	16	0.12	7.083	29.10	71.7	2,012595	284470	166000	44400	37400	23700	12970	58.40	15.60	13.14	8.32	4.54	41.90	11.20	9.43	5.97	3.25	13.00
2230-BJ	12	0.06	6.167	25.00	61.5	1,354230	219510	124400	39100	28800	17700	9510	56.70	17.80	13.10	8.05	4.35	36.60	11.50	8.45	5.19	2.81	12.10
2230-BH	12	0.12	5.167	29.70	70.2	1,415545	274030	149500	46300	39600	25000	13630	54.60	16.90	14.45	9.10	4.95	38.30	11.85	10.14	6.38	3.47	15.50
SECTION III. ANTHRACITE—CHESTNUT SIZE																							
2230-I	16	0.03	17.500	12.57	71.3	2,127309	121550	78600	21100	12500	6770	2580	64.50	17.40	10.20	5.60	2.30	46.00	12.40	7.28	4.00	1.64	6.00
2230-P	16	0.06	11.200	19.57	69.0	2,007070	179100	104700	31350	23000	12950	7100	58.50	17.50	12.50	7.20	4.00	40.30	12.10	8.84	4.97	2.76	9.50
2230-R	16	0.09	9.300	23.50	65.7	1,936722	207650	118000	36300	28100	16450	8800	56.90	17.50	13.50	7.90	4.20	37.40	11.50	8.87	5.19	2.76	11.00
2230-O	16	0.12	8.330	27.00	63.7	1,922694	230780	131300	40200	30800	19500	8980	56.90	17.40	13.90	8.40	3.40	36.23	11.10	8.85	5.35	2.16	12.70
2230-M	12	0.06	7.550	22.90	64.8	1,507080	199260	117000	34750	25450	15430	6850	58.70	17.40	12.75	7.74	3.41	38.00	11.30	8.27	5.02	2.21	10.40
2230-L	12	0.12	5.670	30.10	63.4	1,466224	258700	143000	44400	35200	24100	12000	55.30	17.10	13.60	9.25	4.75	35.10	10.85	8.62	5.87	3.01	12.10
2230-K	8	0.03	5.780	19.55	65.0	997089	172450	105100	31550	19700	11450	4650	61.00	18.30	11.40	6.60	2.70	39.70	11.90	7.41	4.29	1.75	8.40
2230-N	8	0.06	4.500	25.00	61.8	929308	206200	120000	36600	23400	17150	9320	58.00	17.70	11.30	8.30	4.70	35.90	10.94	6.98	5.13	2.91	11.40
2230-Q	8	0.09	3.810	29.70	61.5	938420	245860	136400	43700	32860	20900	12000	55.50	17.80	13.40	8.50	4.80	34.10	10.95	8.24	5.23	2.95	12.80
2230-J	8	0.12	3.250	35.10	59.3	916576	282000	156700	55500	34200	23350	12300	55.70	19.70	12.20	8.30	4.10	33.10	11.70	7.24	4.92	2.43	14.10
SECTION IV. ANTHRACITE—PEA SIZE																							
2230-AN	10	0.03	10.500	13.88	67.6	1,268387	120730	73500	21606	14500	7680	3450	60.90	17.90	12.00	6.30	2.90	41.30	12.10	8.12	4.26	1.96	7.00
2230-AO	10	0.06	7.634	18.82	68.0	1,255086	164470	94400	27950	22600	12900	6620	57.40	17.00	13.75	7.85	4.00	39.00	11.56	9.35	5.34	2.72	8.50
2230-AP	10	0.09	6.750	22.65	65.0	1,278035	189320	105400	32700	26900	15800	8520	55.70	17.25	14.20	8.35	4.50	36.20	11.20	9.25	5.42	2.92	9.75
2230-AQ	10	0.12	5.833	26.20	64.0	1,256709	215400	112000	37600	33300	20200	12300	52.00	17.45	15.50	9.35	5.70	33.20	11.16	9.92	5.98	3.65	11.20
2230-AM	8	0.06	5.983	19.90	66.7	1,022167	170790	97000	29400	23800	13470	7120	56.80	17.20	13.90	7.90	4.20	37.85	11.48	9.27	5.27	2.80	9.30
2230-AL	6	0.06	3.670	25.80	60.5	737963	201040	112600	35200	27800	16200	9240	56.00	17.50	13.80	8.10	4.60	33.90	10.60	8.35	4.90	2.78	9.90
SECTION V. ANTHRACITE—BUCKWHEAT SIZE (INDUCED DRAFT)																							
2230-AV	10	0.06	12.500	10.80	63.7	1,153548	92190	54200	16720	12000	5950	3320	58.80	18.15	13.05	6.40	3.6	37.45	11.56	8.31	4.08	2.29	6.60
2230-AU	10	0.09	10.250	13.50	68.4	1,270337	123820	70100	21750	17700	9550	4720	56.55	17.55	14.40	7.70	3.8	38.7	12.0	9.85	5.27	2.46	8.00
2230-AT	10	0.12	8.330	16.22	66.5	1,204530	144670	80400	25650	21050	11930	5640	55.50	17.75	14.50	8.25	4.0	36.9	11.75	9.64	5.48	2.66	8.30
2230-AS	8	0.06	8.918	12.40	68.0	1,007853	113040	66700	20000	14500	7910	3930	59.00	17.68	12.84	7.00	3.48	40.15	12.00	8.74	4.76	2.37	6.75
2230-AR	6	0.06	6.000	14.87	67.1	801083	133506	73780	25250	18860	10500	5116	55.20	18.90	14.20	7.87	3.83	37.0	12.7	9.53	5.28	2.57	8.50

[a] Loss to dry flue gas.

TABLE 3. (*Continued*)

Test No.	Coal Feed Notches	Air Setting	Duration Hours	Coal Per Hour	Overall Eff.	Total Output Btu	Output, Btu Per Hour						Output, Sq Ft Steam Radiation						Output, Per Cent of Total				
							Total	Sec. 1	Sec. 2	Sec. 3	Sec. 4	Sec. 5	Total	Sec. 1	Sec. 2	Sec. 3	Sec. 4	Sec. 5	Sec. 1	Sec. 2	Sec. 3	Sec. 4	Sec. 5
SECTION VI. ANTHRACITE—BUCKWHEAT—STOKER FIRED (WITHOUT FURNACE BAFFLE)																							
2230-BQ	4	4.50	8	7.65	78.0	636190	79580	30080	26820	11790	6670	4220	331.5	125.0	112.0	49.1	27.8	17.6	37.8	33.8	14.8	8.4	5.2
2230-BP	7	5.20	8	13.81	71.5	1,036510	12[illegible]550	4[illegible]8[illegible]5	40030	20430	11968	7160	539.5	207.5	167.0	85.4	49.8	29.8	38.5	30.95	15.8	9.25	5.5
2230-BO	10	6.35	8	19.60	66.2	1,383550	172940	65730	50570	28390	17790	10460	720.5	274.0	211.0	118.0	74.0	43.5	38.0	29.2	16.4	10.3	6.1
2230-BN	13	6.40	8	24.20	65.7	1,696920	212130	88200	54800	34150	21400	13580	883.1	367.5	228.0	142.0	89.1	56.5	41.5	25.8	16.1	10.1	6.5
SECTION VII. ANTHRACITE—BUCKWHEAT—STOKER FIRED (WITH 21 IN. BAFFLE 10 IN. ABOVE TUYERE)																							
2230-BR	10	6.35	8	18.62	65.6	1,302750	162820	88500	30300	21700	14000	8320	678.8	369.0	126.2	90.5	58.4	34.7	54.4	18.6	13.3	8.6	5.1
2230-BS	13	6.40	8	23.90	68.0	1,732260	216480	122080	39460	28150	18200	8590	900.0	508.0	164.0	117.3	74.9	35.8	56.4	18.2	13.0	8.42	3.98

Test No.	Fuel Bed Depth Inches	Draft	Duration Hours	Coal Per Hour	Overall Eff.	Total Output Btu	Output, Btu Per Hour						Output, Sq Ft Steam Radiation						Output, Per Cent of Total				
							Total	Sec. 1	Sec. 2	Sec. 3	Sec. 4	Sec. 5	Total	Sec. 1	Sec. 2	Sec. 3	Sec. 4	Sec. 5	Sec. 1	Sec. 2	Sec. 3	Sec. 4	Sec. 5
SECTION VIII. BY-PRODUCT COKE																							
2230-Z	16	0.03	6.000	22.80	66.4	1,137383	190340	106000	32500	25900	18620	7820	795.3	442.0	135.0	103.0	77.7	32.6	55.8	17.1	13.2	9.8	4.1
2230-AA	16	0.06	4.800	28.10	61.2	1,071534	223320	120800	38600	31400	20800	11720	930.8	503.0	161.0	131.0	86.8	49.0	54.1	17.3	14.1	9.3	5.2
2230-AB	16	0.09	3.883	34.40	59.5	1,033544	265990	146200	40600	37700	26650	14840	1105.8	610.0	169.0	157.0	111.0	61.8	55.0	15.25	14.15	10.0	5.6
2230-AC	16	0.12	3.467	38.20	61.0	1,049527	302850	166000	46500	44300	28050	18000	1262.0	692.0	193.5	184.5	117.0	75.0	54.9	15.3	14.6	9.3	5.9
2230-AH	12	0.06	3.500	30.00	60.5	825074	236000	131000	39500	33200	21850	10460	982.5	545.0	164.4	138.0	91.5	43.0	55.5	16.7	14.1	9.3	4.4
2230-AG	8	0.06	2.250	33.30	55.4	539318	239700	134000	41300	33600	20600	10200	998.0	558.0	172.0	140.0	85.6	42.4	55.9	17.2	14.0	8.6	4.3
SECTION IX. PETROLEUM COKE																							
2230-U	16	0.03	4.750	24.10	56.5	1,017403	214000	130000	30400	26400	16880	10320	892.0	542.0	126.6	110.0	70.3	43.1	60.8	14.2	12.3	7.9	4.8
2230-T	16	0.06	4.000	27.70	55.0	961233	240310	140500	34900	31350	20070	13490	1001.0	585.0	145.8	130.6	83.6	56.0	58.5	14.5	13.0	8.4	5.6
2230-S	16	0.09	3.734	30.90	54.6	994110	266070	151500	40800	33400	23000	17370	1108.2	631.0	170.0	139.0	95.8	72.4	56.9	15.3	12.6	8.7	6.5
2230-V	16	0.12	3.300	33.70	53.4	935665	283560	167000	38600	35600	26460	15900	1182.0	696.0	161.0	148.4	110.3	66.3	58.9	13.6	12.6	9.3	5.6
2230-W	12	0.06	3.150	25.62	54.0	687667	218230	125000	34030	28700	18500	12000	909.4	521.0	141.8	119.5	77.1	50.0	57.5	15.6	13.1	8.5	5.4
2230-X	8	0.06	2.300	21.40	52.5	408692	177650	95900	28650	25000	17250	10850	740.4	400.0	119.3	104.0	71.9	45.2	54.0	16.1	14.1	9.7	6.1

TABLE 3. *(Continued)*

Test No.	Per Cent Cont.[b]	Duration Hours	Lb Oil Per Hour	Total Output Btu	Output, Btu Per Hour						Output, Sq Ft Steam Radiation						Output, Per Cent of Total				
					Total	Sec. 1	Sec. 2	Sec. 3	Sec. 4	Sec. 5	Total	Sec. 1	Sec. 2	Sec. 3	Sec. 4	Sec. 5	Sec. 1	Sec. 2	Sec. 3	Sec. 4	Sec. 5
SECTION X. OIL BURNER WITHOUT BAFFLE																					
2230-BW	25	8	3.16	364885	45595	23500	9990	6050	3360	1795	190.0	98.0	41.5	29.0	14.0	7.5	51.54	21.9	15.25	7.37	3.94
2230-BV	50	8	6.34	788930	98615	51005	21755	15195	8060	2600	411.5	212.2	90.6	63.3	34.6	10.8	51.8	22.0	15.4	8.16	2.64
2230-BU	75	8	9.47	1,208175	151025	77930	32990	22995	11420	5690	629.2	324.5	137.5	95.9	47.6	23.7	51.63	21.82	15.22	7.56	3.77
2230-BT	100	8	12.53	1,592862	199110	105530	42485	29010	14940	7145	825.1	440.0	172.0	121.0	62.3	29.8	53.0	21.3	14.58	7.5	3.62
SECTION XI. OIL BURNER WITH BAFFLE																					
2230-BX	50	8	6.25	788750	98500	59250	18500	11700	6360	2750	410.8	247	77.0	48.8	26.5	11.5	60.2	18.75	11.82	6.45	2.78
2230-BY	100	8	11.78	1,493177	186540	112820	37270	20360	11130	5100	777.7	470	155.3	84.8	46.4	21.2	60.4	20.0	10.9	5.96	2.74

Test No.	Per Cent Cont.[b]	Duration Hours	Cu Ft Gas Per Hour	Total Output Btu	Output, Btu Per Hour						Output, Sq Ft Steam Radiation						Output, Per Cent of Total				
					Total	Sec. 1	Sec. 2	Sec. 3	Sec. 4	Sec. 5	Total	Sec. 1	Sec. 2	Sec. 3	Sec. 4	Sec. 5	Sec. 1	Sec. 2	Sec. 3	Sec. 4	Sec. 5
SECTION XII. GAS BURNER																					
2230-CC	25	8	107.5	318193	39745	25900	7400	3680	1845	920	165.7	108	30.8	15.4	7.7	3.8	65.2	18.6	9.25	4.64	2.31
2230-CB	50	8	200.0	630085	78764	49700	15050	8300	4340	1374	328.1	207	62.7	34.6	18.1	5.7	[illegible]	19.1	10.53	5.51	1.76
2230-CA	75	8	294.4	943047	117850	75300	21850	11970	6550	2180	491.2	314	91.0	49.8	27.3	9.1	63.9	18.53	10.15	5.55	1.87
2230-BZ	100	8	382.5	1,258233	157300	101000	28450	15750	8700	3400	655.5	421	118.5	65.6	36.2	14.2	64.2	18.1	10.0	5.54	2.16

[b] Per cent of time in operation.

The problem, therefore, consists of extending curves of absorption, such as Figs. 3 and 4, which can then be used as fair estimates of expected additional heat recovery.

While it is seldom desired to remove a section from an existing boiler, the same curves can of course be used as a basis for estimating probable effect.

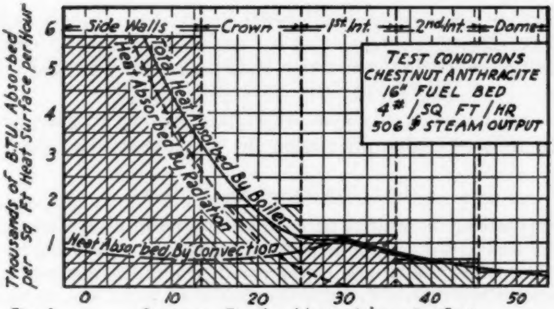

Fig. 3 Square Feet Absorbing Surface

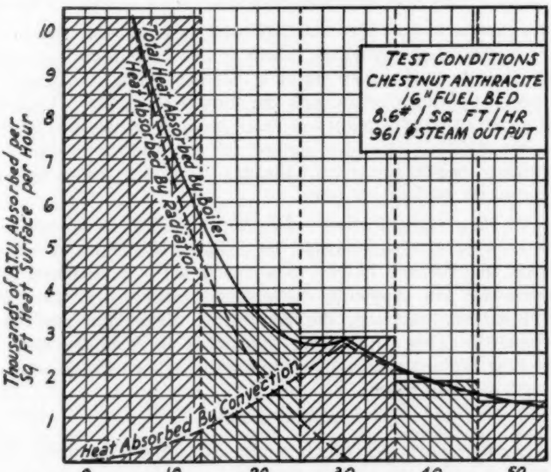

Fig. 4 Square Feet Absorbing Surface

Figs. 3 and 4. Heat Absorbed by Individual Sections Showing Calculated Convection and Radiation for One Typical Test

Reduction of Side Wall Area and Combustion Space

The great majority of anthracite burning domestic boilers are designed in accordance with the requirements of high volatile fuels for large furnace volumes, to insure complete combustion. It has been demonstrated, however, that

anthracite can be burned satisfactorily with practically no combustion space, such as in *Anthracite Industries* horizontal combustion boilers. With anthracite, the chief function of the furnace section is to absorb as much as possible of heat radiated directly from the fuel bed.

The most efficient design for surface-fired or magazine-fed anthracite boilers should therefore include:

1. A firepot of sufficient depth to permit the thickest possible fuel bed consistent with the efficient combustion of the required size, and of sufficient grate area to allow the maximum combustion space consistent with the proper feeding of fuel.

2. Flue passes having a maximum of surface exposed to gases at relatively high velocity without unduly increasing the pressure drop or the size of the unit.

SMALL FURNACES FOR STOKERS

Mention has already been made of the fact that when using anthracite, stoker fired, the side walls are of minor importance as compared with the crown sheet. In view of the sootlessness of anthracite, together with the fact that it is largely a pure carbon rather than a hydrocarbon, it would seem obvious that combustion space over the fire was of minor importance as compared with radiant surfaces directly above the fire, together with well designed flue passes.

This conclusion has in fact been substantiated by a number of tests within the *Anthracite Industries Laboratory,* in which it has been found possible to burn anthracite in an underfeed stoker set at a distance below the crown sheet just sufficient to prevent actual contact with the incandescent fuel. Outputs as high as 350,000 Btu per cu ft of combustion space have been obtained in this laboratory (see report 2057). The ideal stoker furnace therefore should have just sufficient volume to permit lighting of the fire, feeding of ashes into the ashpit, and giving access to replaceable parts.

According to these specifications, the 25 in. boiler in which the relative absorption tests were conducted could be reduced in height by nearly 30 per cent by cutting the furnace height to about 6 in. Or, without altering the external dimensions, but replacing the present unnecessary furnace space by additional intermediate sections, the convection absorption surface could be more than doubled. This would give the stoker used in the tests an output about 10 per cent higher, with correspondingly higher efficiency.

CONCLUSIONS

A number of other variables which might affect the absorption, distribution, and efficiency of a boiler are also of interest. Included in these would be such comprehensive studies as the relation between draft and efficiency; fuel bed depths with various outputs; and the correct design of furnace baffles when using stoker fired or fluid fuels. However, as each of these subjects is of sufficient complexity to warrant a more complete discussion than can be given as a part of this paper, the necessary detailed tests were not included in this investigation. However, in view of the importance of such subjects, together with the scarcity of accurate knowledge concerning these various phases of combustion, it is hoped that other investigators will continue this type of work to provide additional data.

No. 1040

THE DISTRIBUTION OF STEAM IN HEAT TRANSFER SURFACE

By JOHN MCELGIN* (*MEMBER*), PHILADELPHIA, PA.

DURING the past 10 years copper fin and tube heating surface has demonstrated its effectiveness for the transfer of heat from steam to air and is today generally accepted as a universal standard in forced air heating systems. While a wide variety of designs are in existence, the principal differences arise in structural details and the ratios between prime and extended surface. In general, it may be said that available commercial designs have the following characteristics in common:

1. Light weight and compactness.
2. A high condensing rate per unit of weight and face area.
3. Low resistance to air flow.
4. Flexibility in installation permitting the field assembly of individual sections to obtain the desired temperature rise and the air flow area.

These features are, of course, widely recognized and require no further discussion here.

For the most part investigations and developments in the field have been purely on the basis of maximum heat transfer—the principal aim being to arrive at a combination of fin and tube areas, resistance to air flow and a condensing rate that leads to the most effective use of the material employed.

However, the widespread use of automatic temperature control and the reliance that the modern air treating system must necessarily place on such equipment have created a new problem in the design and application of fin and tube surface. It is an outstanding fact that in the usual system the heating surface is only rarely called upon to deliver its full capacity, but during the greater part of the time operates well below the maximum. To fulfill these requirements it is desirable that the heating surface possess not only a high condensing capacity but that it also readily lend itself to systems of continuous control at the lesser demands.

It is the purpose of this paper to describe a means of securing a close coordination between fin and tube steam heating surface and automatic temperature control in all stages of operation. Briefly, this is accomplished by inserting within the usual condensing tubes additional tubes which serve the purpose of distributing steam uniformly along the length of the condensing tube. While

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this simple change would not appear to greatly alter surface characteristics it can readily be demonstrated that steam distributing tubes impart a high degree of flexibility to the usual design. The several features that are described here are the combined result of improved steam distribution and a control system capable of intermediate action.

Automatic control of the heating effect of fin and tube surface or for that matter any type of steam heating element may be obtained in two distinct ways, namely:—(1) by-passing of air and (2) modulation of the steam supply.

A comparison of the two methods on the basis of simplicity and economy indicates advantages for the valve control arrangement. All by-pass dampers and channels are eliminated and the air flows constantly through the heating surface—resulting in a saving in space and first cost and an air volume that is generally unaffected by the thermostatic system. Full control of the surface from maximum capacity to positive shut off is accomplished by a single throttling valve that is subject to proper design and sizing prior to installation.

One of the disadvantages in the by-pass method is the separation of the air handled into hot and cold air streams. Unless a strong mixing action is available on the discharge side of the surface this temperature stratification will frequently persist even in long runs of duct. With streams of air at widely separated temperatures naturally no single point can be selected as representative and close air stream control is a practical impossibility. A typical example of this condition arises where the by-pass principle is used on preheater sections for comfort or process humidifying systems.

If provisions are not made within the condensing tubes to provide for steam distribution, a similar degree of temperature stratification will occur when the supply of steam to fin and tube surface is throttled. When the volume of steam supplied is less than the quantity that can be condensed by the surface, by-passing will be created at the return end. This effect is aggravated by extending the air side area to the practical limit and arises from the high rate at which heat is transferred from steam to metal walls.

The condition may be illustrated by considering a typical example:—Fig. 1 shows a section of surface connected to a steam pressure source of 2 lb per sq in. with the condensate draining to an open receiver through a float and thermostatic trap. This corresponds to the usual arrangement on a pump and receiver atmospheric return system. For the purpose of illustration, the section will be assumed to have the capacity to raise the air at 500 fpm face velocity from 30 to 110 F at a 2 lb steam pressure.

Initially with the valve wide open the condensing tubes will be filled with steam at approximately 218 F. This is assuming an oversized valve and a negligible resistance to flow in the condensing tubes. As the supply valve is throttled a point will be reached where the condensing tubes are just filled with steam at atmospheric pressure at approximately 212 F. A further restriction of the valve will decrease the volume but not the temperature of the steam supplied. The lowest available steam temperature will, in all cases, correspond to the pressure conditions maintained in the return line.

The reduction in capacity effected by the change in steam pressure and temperature conditions within the condensing tubes may be calculated from the relation of steam to entering air temperatures.

$$\text{Reduction in capacity} = 100 - \frac{212\text{-}30}{218\text{-}30} = 3.5 \text{ per cent.}$$

The major reduction in capacity is obtained after the pressure and temperature of the steam have been reduced to the lowest possible level and it is in this range that surface under continuous automatic control will operate the

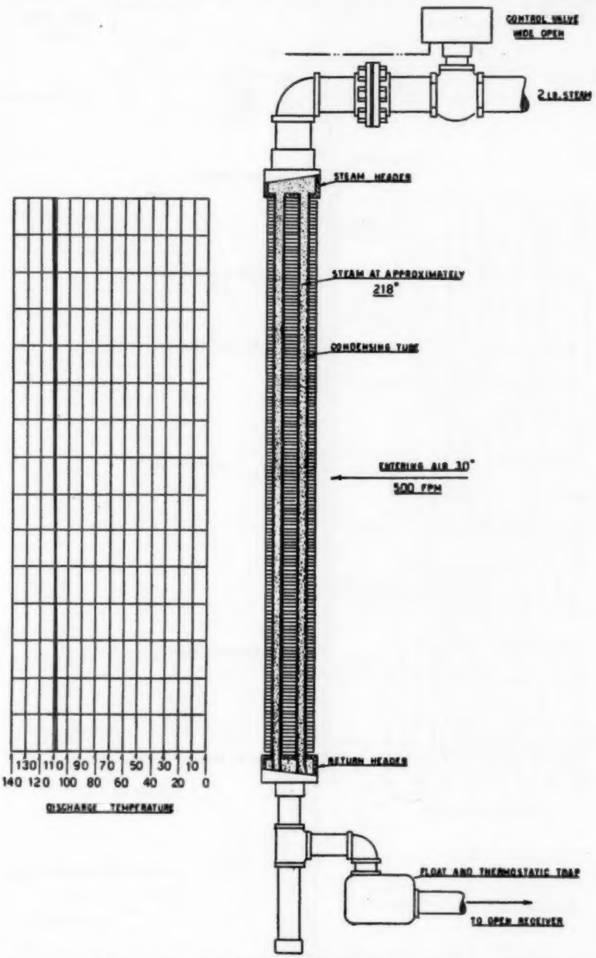

FIG. 1. CHART OF DISCHARGE TEMPERATURES AND SECTION THROUGH HEATING SURFACE WITH STEAM SUPPLY VALVE WIDE OPEN

greater part of the time. As will be apparent from the discussion herewith it is also the region in which the problem of steam distribution becomes a most important consideration.

Continued throttling of the supply valve shown in Fig. 1 eventually produces the condition shown in Fig. 2.

At the steam end the condensing tubes are in full contact with 212 deg steam

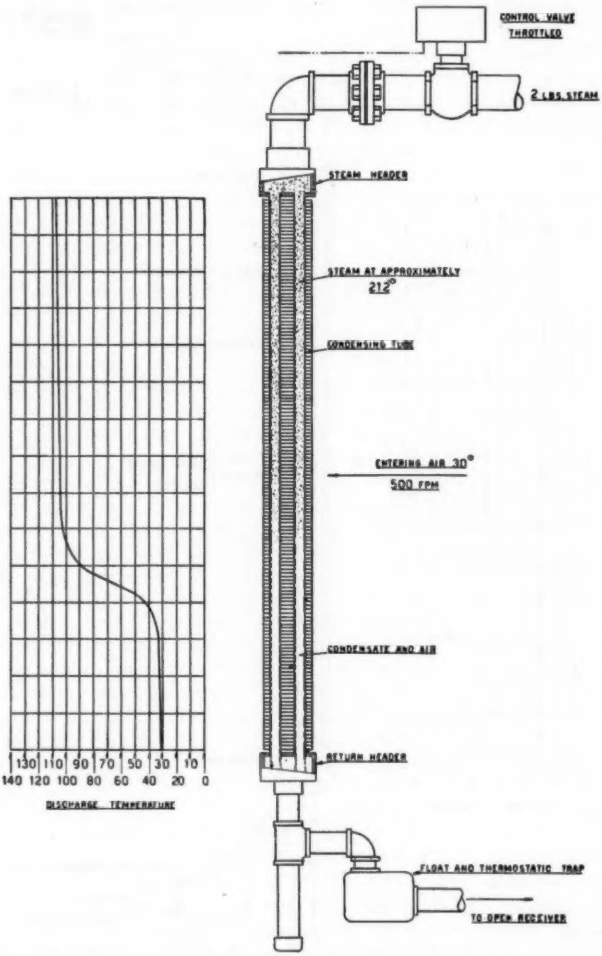

FIG. 2. CHART OF DISCHARGE TEMPERATURES WITH THROTTLING STEAM SUPPLY TO HEATING SURFACE

and that portion will operate at 96.5 per cent of its former capacity. A similar section at the return end will have practically no heating effect since the condensing tubes contact only condensate and air. With the surface under

automatic control the heated portion increases or decreases in accordance with the demand—hence, the comparison with by-passing of air.

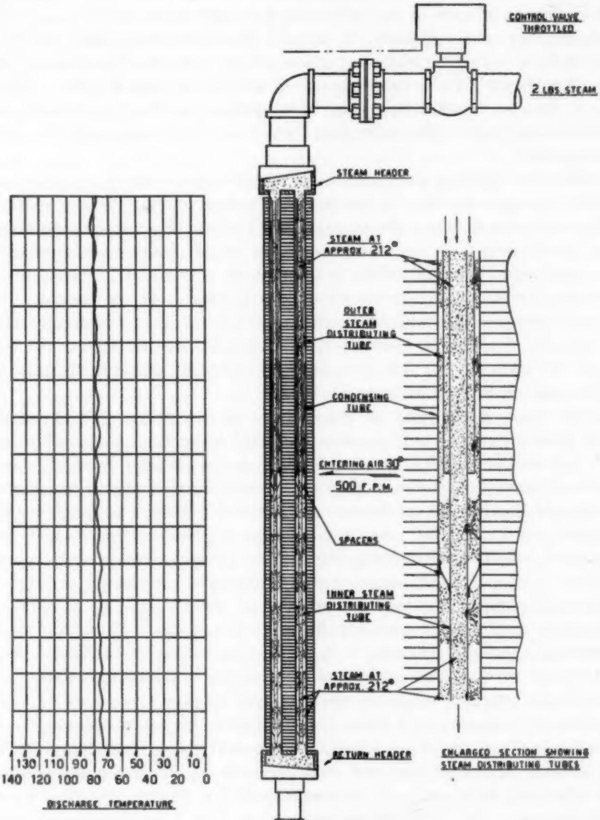

FIG. 3. CHART OF DISCHARGE TEMPERATURES WITH THROTTLING STEAM SUPPLY TO HEATING SURFACE USING CONCENTRIC STEAM DISTRIBUTING TUBES

In view of the danger of freezing the condensate at the return end, continuous valve control is not generally attempted in the presence of sub-freezing air. It is common practice to protect controlled preheaters by additional sections

designed and operated to maintain the entering air above 32 deg. Usually two such sections are used and are controlled by outdoor thermostats adjusted to fully open the respective steam valves at 35 F and at 0 F.

While in the past both of the previous methods have, under many favorable circumstances, proved adequate, it should be recognized that in at least an equal number of cases a wide variation of air stream temperature has been a serious hindrance to the best operation of control equipment. Further the step action necessary on valve controlled preheaters is open to the objection of increased cost and operation that bears only an approximate relation to process demands.

One method of solving the problem of a variable discharge temperature may be obtained through the use of steam distributing tubes. As the name implies, these tubes serve to conduct the steam axially along the condensing tube releasing it at fixed intervals and in practically equal amounts regardless of the quantity supplied. The net effect is to provide a gradual increase or decrease of the whole air stream temperature in accordance with the demand. Coupled with a well designed system of continuous control this arrangement exhibits another equally important characteristic: it permits throttling of the steam supply in the presence of sub-freezing air without danger of tube breakage due to freezing of the condensate.

Naturally, there are limits to the degree of throttling for there exists no method of protection that will prevent freezing when the volume of steam is permitted to fall to abnormally low levels. To derive benefit from this method of steam introduction it is essential that an air stream thermostat be located in the path of the air leaving the preheater section and adjusted to keep the discharge temperature and hence the volume of steam supplied above a safe minimum.

The typical effect of throttling the supply of steam to a section of fin and tube surface equipped with steam distributing tubes is shown in Fig. 3. This is a condition comparable to that previously shown in Fig. 2 and the same steam pressure and temperature conditions will prevail. However, in this case steam entering the supply header is forced to pass first through the distributing tube and thence through the orifices to contact the condensing tubes.

With a limited volume of steam the quantity issuing from each of the orifices is condensed in a small outer tube area opposite the jet, producing a series of active sections along the total length of the surface. This does not, however, create a similar series of hot and cold streams for both the high conductivity of the condensing tube and a close spacing of the outlets equalize the temperature distribution. The distribution shown in Fig. 3 is typical of temperature measurements made in the air stream 6 in. from the face of the surface.

When the steam quantity is increased, the active condensing tube area increases to a point where the tube is filled with steam and the surface is operating at maximum capacity for the entering air and steam conditions prevailing.

A comparison between Figs. 2 and 3 shows that steam distributing tubes are a step towards solving the problem of variable discharge temperatures. It is not the intention, of course, to create the impression that this device is capable of producing absolute uniformity, for this appears from experience to be beyond the limits of practice. It is possible, however, by careful regulation of the orifice sizes and inner tube areas to maintain a comparatively negligible difference between any two points in the air stream.

As an example, the type of two-row section illustrated in Fig. 3 will have the capacity to heat air from 30 to 110 F at 500 fpm and a 2 lb steam pressure. With the supply valve throttled, the average variation in discharge temperature over the face of the section with uniform air flow will vary over a range from 5 to 10 F, depending upon the length of the condensing tube. In direct contrast the same section without distributing tubes, will in some cases under less than maximum demand conditions separate the air volume into air streams that may vary approximately 70 F apart over the entire surface.

The method used in the design of distributing tubes is largely one of trial and error based on tests of the particular section at hand. As would be expected the major controlling factors are the finned length and the diameter of the condensing tubes. It is obvious that long sections require more careful attention than short ones of equivalent fin spacing and capacity per unit of face area. The diameter or the internal area of the condensing tube directly affects the space available for steam distribution and therefore limits the maximum length of finned tubing. In general, surface designs employing condensing tubes ⅝ in. or over are more readily adapted to the principle of steam distribution than ones with smaller tubing.

The original design of distributing tube was based on the use of a single tube extending the full length of the condensing tube. This construction was found to have several disadvantages chief among which was the practical limitation of finned tubing for good distribution to approximately 3 ft. An arrangement adapted to lengths up to 6 ft with ¾ in. condensing tubes and capable of much better steam distribution was that of employing multiple tubes all of which receive steam directly from the primary header. The design shown in Fig. 3 employs two concentric tubes of 9/16 in. and ⅜ in. diameter inside a ¾ in. condensing tube. The first or larger tube serves to supply one half of the surface and the second tube the remainder.

A feature of this construction is that the division of flow occurs at the header and the relative quantities supply both ends if subjected to close control.

From a control standpoint the practical importance of the more uniform air stream temperature obtained with the distributing tubes is dependent upon the location and the function of the heating surface. Some applications are naturally more sensitive in this respect than others, the most critical being those where close control is desired from a thermostat located directly in the path of the air leaving the surface, and there is no opportunity for even partial mixture of the streams. An example of this condition is in preheater sections for humidifying systems where the proper location of the dewpoint thermostat has frequently been a problem. Booster heaters fall under much the same heading. An arrangement less sensitive to temperature stratification is that of mixing the air in a single fan. It should be noted, however, that passage through a fan does not always produce a complete mixture since this will depend upon the temperature difference and the location of the surface with respect to the fan inlet or inlets. Many unit designs employ multiple fans with the surface under steam control and the condensing tubes parallel to the shaft axis. In this case, practically no benefit is derived from mixing and the fans handle air at widely different temperatures. Here and in similar instances steam distributing tubes will usually be found to be of value in reducing the temperature stratification to a minimum.

Perhaps the outstanding feature of this device as applied to fin and tube

heating surface is the inherent protection offered against freezing when the supply of steam is throttled. This arises directly from the fact that the reduced volume of steam is brought into contact with a series of areas distributed along

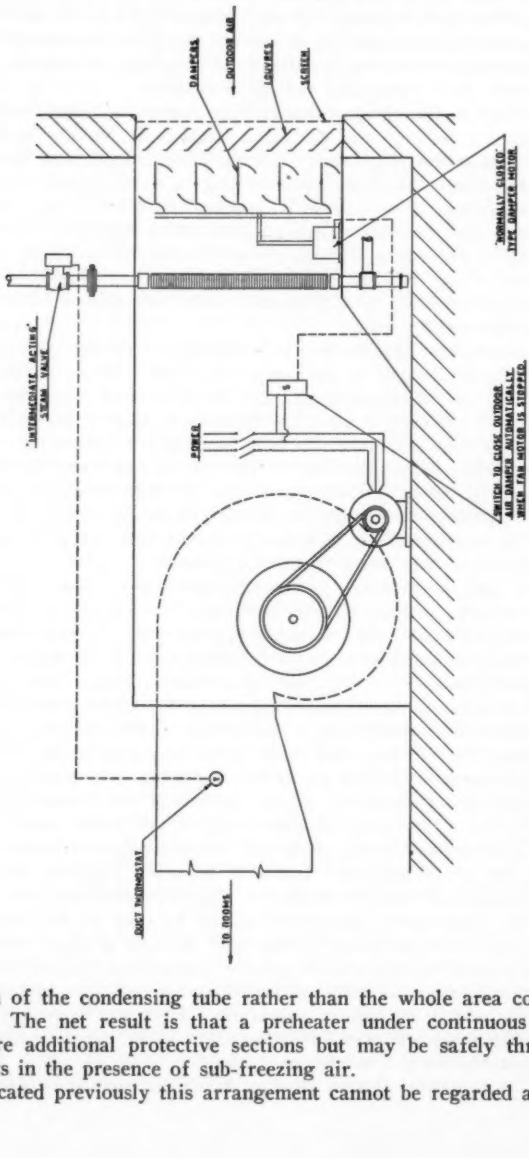

FIG. 4. GENERAL ARRANGEMENT OF FAN AND HEATING SURFACE USING STEAM DISTRIBUTING TUBES

the length of the condensing tube rather than the whole area concentrated at one end. The net result is that a preheater under continuous control does not require additional protective sections but may be safely throttled within wide limits in the presence of sub-freezing air.

As indicated previously this arrangement cannot be regarded as a universal

cure for any and all freezing problems but will have advantages during operating periods where the steam supply is continuous and where a low limit thermostat adjusted for 45 F or above is employed on the discharge side of the surface. Actual protection is then accomplished in the automatic control equipment and the distributing tube simply insures that all parts of the condensing tube will receive a representative quantity of steam. It should be observed that with the discharge temperature fixed safely above the freezing point the demand for steam increases as the outside temperature falls and the surface approaches towards maximum capacity. This automatically provides against extremely close throttling in cold weather. The use of a low limit thermostat does not impose a special control requirement, for this type of instrument usually forms an important part of every automatically controlled ventilation system.

Fig. 4 illustrates a simplified arrangement possible when a heating surface equipped with distributing tubes is used to preheat outdoor air. While this is shown for the usual tempered air ventilating system it serves as an example for other preheater applications since the only change required is the relocation of the duct thermostat. It should be noted in this arrangement that the usual positive acting preheaters are omitted and full control is accomplished with the thermostat located in the leaving air stream. Normally a two-row section of the average design will meet this requirement but where an additional section is necessary this should be separately valved and operated in sequence with the first section.

Throughout the previous discussion use has been made of the term *continuous steam control.* By this is meant a combination of a steam valve and a thermostat so selected that the supply of steam is exactly the right quantity to meet the thermostatic demands without hunting. In view of the fact that this type of operation is essential to the best performance of any air heating system it will be well to observe the conditions under which it is possible.

The control that has been found most suitable for steam valve operation is of the intermediate acting or proportioning type in which the valve assumes a fixed position for various temperatures affecting the thermostat. The temperature change required to cause full travel of the valve is termed the *thermostat differential.*

The use of proportioning control in principle is, however, never sufficient to insure continuous results in the field. This will require that careful attention be given: (*a*) thermostatic differentials; (*b*) valve design and size; and (*c*) operating steam pressures. All three factors are very closely related and should be considered in combination to obtain the desired results.

In general, widening of the thermostatic differential will tend toward more gradual control and at the same time a system less sensitive to changes in demand. In view of the latter it is desirable to employ as narrow a differential as possible although a slight change in control point with the demand must always be accepted if hunting is to be avoided. Normally, room type thermostats have a differential of 3 to 6 F whereas the differential on duct type thermostats is three or four times this value. This does not imply a lowering of the control point by this amount, for a wide open valve may not be necessary to meet the maximum demand. This will depend on the valve size and the working steam pressure since a reduction in either will produce much the same result as widening of thermostat differential. The usual shift in the control

point for outdoor air preheater applications falls in the range from 3 to 6 F differential.

Attempts to reduce this by increased valve size or decreased ductstat differentials generally result in hunting. A practical means of securing as close a differential as possible is by employing instruments whose differentials are capable of readjustment at the job.

To function properly under close automatic control a valve must offer considerable resistance to the flow of steam even in the wide open position. This is verified by the fact that the major reduction in surface capacity is obtained after the pressure in the condensing tubes has been reduced to a point near the return line pressure which may be either atmospheric or vacuum. With the surface under close control the pressure drop through the valve is practically equivalent to the differential from steam main to return main excepting, of course, the slight resistance to flow in the condensing tubes. If the valve is oversized, and this appears to be the usual tendency, control over the surface capacity is concentrated in a small percentage of the total movement causing the system to be unstable. To some extent an oversized valve can be compensated for by increasing the thermostat differential, but the better method is to employ a smaller valve particularly if two valves are to be operated in sequence from one instrument.

An effective method of sizing control valves is on the basis of resistance to flow of steam under maximum conditions and with recognition of the line pressure employed. For example, on a 2 lb steam pressure system wide open, a valve having a resistance under maximum demand equal to from 50 to 75 per cent of the initial pressure will give good results. As the initial steam pressure increases to 10 lb the pressure drop through the valve should fall in the range 70 to 80 per cent of the initial pressure. Higher line pressures will require correspondingly greater pressure drops. The principal consideration is to have the valve as close to the major control range as possible so that any valve movement will produce a corresponding change in air stream temperature.

Some attention should also be given to the design of control valves for modulated results with the view of obtaining an approach to straight line lift-flow characteristics. This may be approximated through the use of globe valves equipped with V ports or other similar throttling devices. The tendency of a standard flat disc valve to have its control range concentrated in a small initial lift renders it generally unsuited for this work.

As may be inferred from the above explanation, the matter of working pressures is intimately connected with that of valve sizing. Experience has proven that the most practical range of steam pressures for close control is 2 to 10 lb. This is because the heating surface under steam control operates in a low pressure region and the valve must function with the necessary differential. Where high pressure steam is employed the problem of sizing becomes quite critical and careful attention must be given the erosion resisting qualities of the disc and seat materials.

SUMMARY

1. Several of the difficulties encountered in the application of automatic temperature control to fin and tube steam heating surface are a direct result of the temperature stratification created when air is by-passed or when the

steam supply is modulated. In the first the condition is generally fundamental to the process, while in the second it is due to complete condensation in a limited part of the section nearest the supply.

2. In a system employing fin and tube surface under valve control it was found that temperature stratification may be reduced by inserting steam distribution tubes within the usual condensing tubes. These tubes serve to distribute reduced volumes of steam over a maximum length of condensing tube and so effect a practically uniform discharge temperature.

3. This arrangement inherently resists freezing of the condensate in the heating surface. The steam supply may be throttled within wide limits in the presence of sub-freezing air, making it possible to eliminate the positive acting preheaters usually installed to protect the controlled preheater. The conditions under which this operation is possible are simply that the steam supply be continuous and that the volume be maintained above a low limit by an air stream thermostat installed in the path of the discharged air.

4. Any system of proportioning steam valve control as applied to fin and tube surface may be considerably enhanced by taking the proper precautions to prevent *hunting*. Aside from the type of control the factors that determine whether or not the operation is free from this defect are: (*a*) thermostat differential; (*b*) valve size; and (*c*) operating steam pressures. Recognition of their influence on the control system will materially improve the final results.

DISCUSSION

H. F. Hutzel (Written): Among the advantages claimed for a heating coil with steam distributing tubes, described in this paper, is that when applied to a modulating controlled system it prevents temperature stratification due to the fact that steam is uniformly distributed over the entire face area of the coil. It seems to me that the author has over-emphasized the disadvantage in by-passing air around the heating coil. No doubt if air temperatures are observed in a duct system immediately beyond a heating coil and air mixing chamber, temperature stratification would be noticeable. It is, however, inconceivable that with conventional air velocities, turns and grilles, which are common to the average duct heating system, and all of which create turbulence, that stratification should be noticeable at the point of air delivery.

It appears to me that a heating coil with steam distributing tubes would offer a distinct advantage in its application as a tempering coil on account of the elimination of danger of freezing which so often occurs in a coil with entering air below freezing temperature. I should like to ask whether there is not danger in throttling a coil of this type below a minimum opening beyond which the volume of steam delivered would not be sufficient to uniformly distribute steam to all the openings in the distributing tubes. Also I would like to ask the author if he has had any experience with the coil he describes when connected with a float trap for condensate return and the conventional air valve for venting. It is conceivable that steam might reach the air valve before the coil is entirely relieved of the air. In this case coil efficiency would be materially impaired.

F. B. Rowley (Written): This paper deals with an important phase of many problems in air conditioning work. As the author states, it is difficult to get good steam distribution throughout the tubes of finned surface heaters when throttling the supply of steam. The author's discussion of the use of steam distributing tubes is interesting and supports the conclusions in the summary. Some actual data would have been a valuable contribution to the paper, as there are naturally differences of opinion as to the degree in which the various conclusions may apply.

It seems reasonable that the distribution system described would improve the temperature distribution, but the extent of its improvement would be more convincing if test data were supplied. For instance in Fig. 3 the diagram at the left would indicate that the temperature of the air coming from the heater varied through narrow limits from top to bottom, but this is evidently not a test curve, as the variations in temperature do not correspond to the opening shown in the steam distribution tubes at the right of Fig. 3. Likewise the curve of Fig. 2 appears to be one showing the general characteristics to be expected rather than one produced from actual test data.

In paragraph two of the summary the author states that stratification of temperature may be reduced, but does not state to what degree the temperature stratification was reduced, and in paragraph three he states that the arrangement inherently resists freezing of the condensate, etc. It would seem that for temperatures of —20 F it might be a dangerous procedure to omit the positive acting preheating coils. These points would be more convincing if test data were supplied.

JOHN MCELGIN: With regard to Mr. Hutzel's first point; stratification created at the heating surface will exert its greatest influence on control instruments located in the path of the air leaving the surface or upon a second process such as humidifying that takes place directly after the incoming air is preheated. In long runs of duct this stratification will unquestionably disappear by the time the point of delivery is reached, but this does not lessen its influence as far as control by a duct thermostat is concerned. To function accurately such instruments must either be subjected to a uniformly mixed air stream or a portion of the air volume that is representative of the total. Where wide temperature stratification exists the control point of the instrument is obviously not a true indication of the air stream temperature.

Mr. Hutzel's second question refers to the danger of freezing if the steam supply is reduced to very low levels in the presence of sub-freezing air. In this connection it should be noted that the steam distributing tubes only insure a representative quantity of steam to all parts of the condensing tubes and external means must be employed to prevent this supply from falling below a safe minimum. This protection is provided by a duct thermostat located in the path of the air leaving the surface and adjusted for a control point of 45 F or more. Usually the control point is in the range of 60 to 70 F. Thus, as the outside temperature progressively falls the quantity of steam required to maintain the minimum point increases and the surface approaches its maximum capacity.

This type of surface has been in use for several years with returns variously connected to thermostatic float and thermostatic and float traps with automatic air valves with no evidence of air binding. Since the path the steam must take in initially filling the condensing tubes and the path the air must take in being exhausted is the same, it does not seem reasonable that this condition could occur.

The answer to Professor Rowley's first question regarding the degree of reduction in temperature stratification is essentially answered in the body of the paper. For example, the section illustrated in Fig. 3 is capable of raising the air temperature from 30 to 110 F with a 500 rpm velocity. For stages of valve throttling from 20 per cent upward, the average variation in temperature may be held to 10 F if distributing tubes are used. This is comparable to the 70 F difference that would exist with no tubes or a larger difference if face and bypass dampers are employed. While no reference is made in the test to this matter, Figs. 1, 2 and 3 have been drawn from actual test data under the conditions indicated.

Professor Rowley's second question also refers to the danger of freezing in the presence of air below the freezing point and the answer to Mr. Hutzel's second question will clarify this point. Actually at an entering air temperature of —20 F and below, a section of surface selected for practical conditions would be operating at or near full capacity and would be almost filled with steam. Hence the conditions would be the same as if a positive acting preheater had been used.

No. 1041

PROGRESS IN AIR CONDITIONING IN THE LAST QUARTER CENTURY

By W. H. Carrier * (*Member*), Newark, N. J.

OUTSIDE of the engineering profession, which has been interested for many years in the problems of ventilation, air conditioning today is considered a new art. That it is not a new art is sufficiently attested to by the fact that just a quarter of a century ago it had been sufficiently developed from an engineering and commercial standpoint to attract the attention of the Program Committee of the *American Society of Mechanical Engineers.* So, to the writer's surprise, an urgent request was received from that Society to prepare and present a paper on Air Conditioning. The date of this request was the Spring of 1911.

Development of the Art Prior to 1911

At this time there had been organized at least three concerns, wholly or partly, in the air conditioning field, S. W. Cramer, Charlotte, N. C., in the textile field, Warren Webster & Co., Camden, N. J., who had started an air conditioning department, and the Carrier Air Conditioning Co. of America, incorporated in 1908 as a subsidiary of the Buffalo Forge Co. In addition, there were also a few contractors who had become interested in air conditioning and had made some installations. Most notable among these was W. L. Fleisher, who has since contributed considerably to the art.

That date probably marks the first recognition of air conditioning as a distinct art in the engineering profession, while the present date, exactly 25 years later, properly may be chosen as the date of the first general and recognized public acceptance of air conditioning as an essential factor among the conveniences of our modern material civilization. Not only does this acceptance now prevail among all classes in America, but it is also most assuredly reached in all parts of the civilized world, especially in the countries which the writer had occasion to visit recently, South Africa, India, Japan, Australia, as well as Europe.

Therefore it may not seem amiss, at this particular time, first to outline the state of the art at the time of the first engineering presentation 25 years ago, second, the history of the developments that led up to the existing status and,

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* Chairman of the Board, Carrier Engineering Corp.

Presented at the Semi-Annual Meeting of the American Society of Heating and Ventilating Engineers at the Joint Session with the *A. S. R. E.*, June, 1936, Buck Hill Falls, Pa.

third, and more especially, to point out the significance of certain vital developments which have made possible its recent general public acceptance.

As a result of the request from the *American Society of Mechanical Engineers* for a paper on air conditioning, a treatise was prepared by the author entitled, Rational Psychrometric Formulae,—Their Relation to the Problems of Meteorology and of Air Conditioning. Since an intensive study had been made by the writer on this subject during the previous 9 years, it was only necessary to assemble and arrange experimental facts and mathematical deductions to put this material in shape for presentation. However, upon the submission of this paper, the Program Committee immediately added a request for a companion paper to be presented at the same meeting in December, 1911. This was to give a picture of the practical side of air conditioning, as well as its more theoretical, mathematical and physical aspect. A compliance with this request was not difficult owing to the availability of previously prepared data. This companion presentation was entitled, Air Conditioning Apparatus—Principles Governing Its Application and Operation.

Assistance was given in the preparation of this paper by F. L. Busey. These two papers presented a very complete picture of both the theoretical and practical aspects of the art as then practised by the Carrier Air Conditioning Co. Not enough was known, at that date, about the then competitive systems to permit their inclusion. The Cramer system was applied, at that time, almost exclusively to textile mills with atomizing spray units distributed throughout the mill with a smaller number arranged to take all or a portion of outside air. His control system originally used wet- and dry-bulb thermometers actuating an electric current. His later and greatly improved form of mechanically operated hygrostat had just been introduced.

As a basis for clearer understanding of the state of the art at the time these papers were presented 25 years ago, a brief history of its development will be interesting.

While the term *air conditioning* has become almost a household word, and so far, there seems to be no other substitute which will simply convey the same idea, its origin is not well known. Although the term *air conditioning* was employed as a part of the name of the Carrier Air Conditioning Co., the first company to broadly enter this field, it was actually originated by S. W. Cramer of Charlotte, N. C., whose name has already been mentioned. About 1907 Mr. Cramer presented a paper before the *National Cotton Manufacturers Association*, on his system of humidification and humidity control for textile mills. He showed how the moisture content of a product followed the moisture content of the air, referring to work done by Schlessinger of France and others in this field of study. From this he argued that the control of the moisture in the air would necessarily control the moisture content in the product. The measurement and control of the moisture content in textiles was then generally known in the trade as *conditioning* and he proposed the logical term of *air conditioning* for the means which would maintain a desired humidity in the room where textiles were processed. Thus, the term of *air conditioning* applied originally only to the control of the moisture content of the air in reference to its effect on hygroscopic materials. In the adoption of the term the following year, in the name of the Carrier Air Conditioning Co., it was intended to have a broader application than this and to include, beside humidity

control, air cooling, heating and cleaning, as well as the general control of ventilation. Since that time others have attempted to broaden or to use the term more loosely, as for example, where a disc fan is employed to move the air, or where a warm air furnace is provided with a fan and a moistening means; however, although called air *conditioning,* these last named applications obviously are not in accordance with any true conception of the term.

Like most industries, air conditioning developed as a result of a realized but unfilled need. Also as in many industries, other developments have been greatly accelerated by the fortuitous discovery of certain basic principles which could be applied advantageously to the improvement of the art.

The author's attention was first directed to the possibilities in this field by a late member and past president of this Society, W. S. Timmis, consulting engineer, of New York City. Mr. Timmis had been retained by a Brooklyn lithographer to design, among other things, a heating and ventilating system for his plant. The lithographer had told Mr. Timmis of his difficulty with humidity, both in winter and summer, and Mr. Timmis had made some study of the possibilities of dehumidifying in summer by use of liquid calcium chloride, as well as by use of condensing coils. It was recognized by Mr. Timmins that nowhere was there any equipment manufactured for such a purpose, and, through J. I. Lyle, a request was made to the Buffalo Forge Co. to undertake a series of experiments relating to the possibility of dehumidification of the air in summer by one or both of these methods.

Since the author, who was then an employee of the Buffalo Forge Co., had just completed his first self-assigned research work on heat transmission in air heaters, he was, naturally, assigned to this project. These experiments were conducted through the summer of 1902 and in the early part of 1903. Aside from some interesting data on greatly increased factors of conductivity of cooling coils in moist air and upon the practical absorption rates of concentrated calcium chloride solution, little practical result was accomplished. However, specific information of the existing data on atmospheric moisture was obtained principally from the Weather Bureau's psychrometric tables and consideration was also directed to the fact that when calcium chloride, or any other substance, absorbed moisture out of the air an exactly corresponding amount of latent heat was released in the form of sensible heat. This phenomenon, submitted in the report at that time, was carefully analyzed. On account of its heating effect and for other reasons, the use of calcium chloride, as a dehumidifying agent, was not considered practical. However, the observation of this *one* phenomenon led to a train of thought, which eventually was to become important. This experiment disclosed the inter-relation of latent and sensible heat in the air when its moisture content was altered without the addition or subtraction of external heat. It also led to complementary experiments upon the process of evaporation of water into air and, finally, into the development of the principles upon which air conditioning was founded as presented in the paper entitled, Rational Psychrometric Formulae, previously referred to.

It also led to a further study of the need for devising suitable equipment for carrying out air conditioning processes as well as to thought upon the need of various industries for maintaining atmospheric conditions, independently of external weather variations. So, in the winter of 1903 and 1904, a spray type of air conditioner was finally devised, suitable for such purpose and with

definite means of controlling absolute moisture content of the air leaving the equipment. Thus, it became possible to definitely control the relative humidity within an enclosure. Little thought, however, was given at that time to adapting this same process to the absorption of heat generated within an enclosure as the necessary and complementary step of true air conditioning. It was not until 1906 that the discovery was made of the necessary relationship between quantity of saturated air supply and the amount of heat generated within an enclosure.

This disclosure came as a result of the study of the first application of such a system to a cotton mill located near Charlotte, N. C. Like many discoveries,

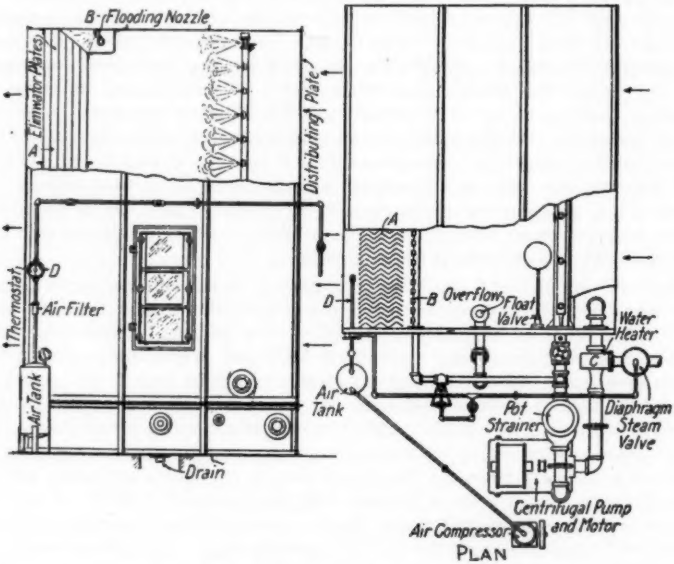

FIG. 1. AIR WASHER WITH HUMIDITY CONTROL

this, when found, would have seemed quite obvious from the beginning. Actually it was not. In this study an interesting fortuitous relationship was discovered between the cooling capacity of saturated air and the relative humidity which could be maintained with varying temperature, that is, that *the differential between the dew point of air introduced and the temperature of the room was practically constant for any relative humidity irrespective of the variation in basic temperature.* For a fixed room temperature, of course, this was obvious and was the foundation upon which the basic patent for the dew point method of controlling relative humidity was obtained.

This spray type of air conditioner, as shown in Fig. 1, with certain modifications, was also used about this time for dehumidification by artificially cooling the spray water and for humidification in winter by heating of the spray water.

The general design and features of this early spray type air conditioner are now quite generally adopted throughout the world and there has been little or no change in the design of its essential features in the past 30 years.

Air washers for cleaning air had been designed and had come into general use prior to this time. There were three pioneers in the field of air washers whose names should be mentioned, one of which was Joseph McCreery, who first designed an *S* type of air washer with sprays and eliminators for use in ventilation equipment on boats for the Great Lakes. The principal air washer built at this time, was that designed by Mr. Thomas, of Thomas and Smith at Chicago. This air washer did an excellent job of air cleaning and elimination of free moisture, but the type of sprays employed was not suitable where an exact control of the moisture content was required. There was also a combination of fan and air washer manufactured by Zelwieger of St. Louis, which seemed, for a time, to have some meritorious features, particularly that of space saving.

The types of eliminators employed at this time, however, while efficient in air washing, were not adequate for proper separation of a finely divided spray such as proved to be required in air conditioning.

The method of spray production and distribution was also a matter of concern. Some type of nozzle in which the water was distributed and broken up into fine particles by means of centrifugal action was desired for this function. A simple type of nozzle which had already been developed was selected, which operated by hydraulic action, producing a rotating stream of water through the orifice which, upon issuance, burst into fine particles spread over a wide area. There are two or three methods of producing this rotation, all of which are now in use. The principal requirements in design are simplicity, freedom from clogging, and ease of cleaning.

The general type of nozzle first adopted over 30 years ago, is now the one most generally employed. This type of nozzle was actually invented by a horticulturalist for the purpose of spraying insecticide and supplants older designs which were found unsatisfactory. This is an interesting illustration of how one art may borrow, with profit, from another.

Following the development of satisfactory equipment and processes of controlling the humidity of air, together with the discovery and analysis of the physical laws involved, there was a rapid development in application to industry starting first with textile mills. Here the problem was always one of increasing, as well as controlling relative humidity combined with cooling to remove the large amount of heat generated by the textile machinery.

To quote from the introductory paragraph in the paper Rational Psychrometric Formulae: "The application of this new art to many varied industries has been demonstrated to be of greatest economic importance. . . . In many other industries, such as lithographing, the manufacture of candy, bread, high explosives and photographic films, and the drying and preparing of delicate hygroscopic materials, such as macaroni and tobacco, the question of humidity is equally important." Mention was also made in this publication of the desirability of *application of air conditioning to mines.* This will indicate the commercial status of air conditioning 25 years ago.

In this paper a new theory of psychrometry was also outlined, based on the method of determining the temperature of adiabatic saturation. The entire

theory was embodied in four significant and four basic laws or principles as follows:

"a. When dry air is saturated adiabatically the temperature is reduced as the absolute humidity is increased, and the decrease of sensible heat is exactly equal to the simultaneous increase in latent heat due to evaporation.

"b. As the moisture content of air is increased adiabatically the temperature is reduced simultaneously until the vapor pressure corresponds to the temperature, when no further heat metamorphosis is possible. This ultimate temperature may be termed the temperature of adiabatic saturation.

"c. When an insulated body of water is permitted to evaporate freely in the air, it assumes the temperature of adiabatic saturation of that air and is unaffected by convection; *i.e.*, the true wet-bulb temperature of air is identical with its temperature of adiabatic saturation.

"From these three fundamental principles there may be deduced a fourth:

"d. The true wet-bulb temperature of the air depends entirely on the total of the sensible and latent heat in the air and is independent of their relative proportions. In other words, the wet-bulb temperature of the air is constant, providing the total heat of the air is constant."

The heat content or total heat of the air was calculated in this paper and was shown to be determined by the wet-bulb temperature. A chart known as the Psychrometric Chart was devised which presented the relationship of wet- and dry-bulb temperature, humidity and total heat. This chart has since formed the basis for all air conditioning calculations, and has permitted an easy solution of all air conditioning psychrometric and drying problems on the assumption of a standard barometric pressure. Formulae were developed which permitted the construction of similar charts for other barometric pressures or the calculation of special relations not embodied in the original chart.

It will be noted that for all practical purposes that these original laws are generally accepted today for engineering purposes. It should be observed, however, that it has since been shown that there is a slight error in the assumption that the wet-bulb temperature is identical with the temperature of adiabatic saturation. Only one error was originally considered, and that was the error due to radiation. It has since been shown, however, that there is another compensating error, which, for practical purposes, is so slight as to be negligible. This departure is caused by a greater rate of diffusion of water vapor than that of air, so that theoretically it is possible to have a wet-bulb temperature lower than with the temperature of adiabatic saturation. This discrepancy is so small, with water evaporating into air, that it is noticeable only with the most refined thermometer readings. Actually, as it tends to counteract radiation of air, the wet-bulb temperature reading coincides almost exactly for all practical purposes with the true theoretical temperature of adiabatic saturation. This is a fortuitous relationship existing with water and air but does not hold as well with other combinations. This discovery is the principal contribution which has been made in the theoretical field since the time of the publication of the first paper. A further study of the radiation error of the wet-bulb thermometer was also presented by the author, with D. C. Lindsay, at the Annual Meeting of the *American Society of Mechanical Engineers* in 1924. This discussion gives a detailed account of further research done subsequently in this field. The important point, however, is that for engineering purposes in the field of air conditioning, the relationships given in 1911 are still basic and accurate.

In the design and manufacture of air conditioning apparatus there has been considerable advancement, as might be expected. The early status of the art

in reference to apparatus and its application is quite fully given in the second paper Air Conditioning Apparatus.

The construction of the humidifier and the dehumidifier is still arranged substantially as used in the original design. Improvements occurred primarily in the simplification of control equipment and in methods of water cooling for dehumidification or refrigeration. The *dew point* method of control, however, probably remains as important as at the date of the first publication. It is now in general use principally because it is a basic air conditioning principle.

A considerable portion of the paper Air Conditioning Apparatus was devoted to a mathematical analysis of experimental data on heat transfer which is important in heating, ventilating and air conditioning fields. It is believed that this was the first engineering publication of the theory of heat transfer between a surface and dry or moist air at varying velocities. Some minor corrections would be made if this theory were to be presented today, but in general, time has proven it most useful and accurate.

At the time of writing this paper, but little use was made of cooling coils or cooling surfaces in air conditioning and in reality this method has played a part of little importance until the last few years. Today its increasing commercial importance has been made possible by two great advancements, first, in improved surfaces and second, the introduction within the last 5 years of new refrigerants. These developments will be discussed later.

In the 1911 paper there is complete discussion of the theory of heat transfer and of air cooling and dehumidifying with cooling coils. This same theory is in general use today in determining intricate calculations. The relationship of the increment of latent heat per pound of air to increment of sensible heat per pound of air is exactly the same theory as employed today. It also shows the by-pass effect of coil surface and the correct method to be employed in calculation of the relative humidity, the moisture content and temperature of the air leaving the coil surface and emphasizes the discovery that the air is not necessarily saturated. These subjects are discussed in this paper under the headings: Air Cooling and Dehumidifying with Cooling Coils, Rate of Transmission Between Air and Water Where Condensation Occurs, Moisture Content of Air Leaving Surface Dehumidifier, and Theory of Convection with Forced Circulation.

As this publication has been out of print several years, it is much less well known to the engineering profession than the first paper, but in many respects it has nearly equal engineering value today, especially in the light of recent developments toward unitary air conditioning equipment with surface cooling.

Up to 1911 and for 10 years thereafter, air conditioning was employed almost entirely in industry. It was not realized previously the tremendous effect it was to have later as applied to requirements of human comfort and particularly in summer cooling in connection with refrigeration. For this reason the public has heard little of it until recent years.

This was true in spite of the fact that the author wrote a catalog in 1905 on air washers for the Buffalo Forge Co. in which the advantages of application of this type of air washer to cooling and dehumidifying processes were emphasized and it was predicted that it would be used in the cooling of churches, theaters, hospitals, etc., in the near future. This prediction, while it has been

since proven essentially valid as to method, was entirely wrong as to estimate of time required for public acceptance.

Reference should be made to an interesting paper on Early Comfort Cooling Plants by G. R. Ohmes and A. K. Ohmes in the June 1936 issue of *Heating, Piping and Air Conditioning.* These attempts at comfort cooling range from the use of ice, as early as 1880, to a mechanical refrigeration installation reported in 1898.

The earliest comfort cooling installation using mechanical refrigeration of which the author has heard is that cited at a recent meeting of the *Refrigerating Machinery Association* by C. W. Vollmann, president, Linde Canadian Refrigeration Co., Ltd., Montreal. Mr. Vollmann tells of an installation made by the English Linde Co. in about 1887 using ammonia refrigeration to cool a Rajah's palace in India.

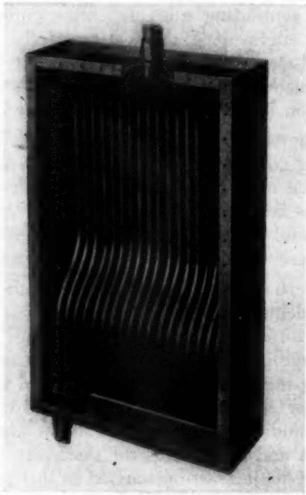

FIG. 2. IMPROVED HEAT TRANSMISSION SURFACE FOR HEATING AND COOLING

IMPROVEMENTS IN THE ART SINCE 1911

Having discussed the status of the art 25 years ago in relation to present practices, it is well to survey and analyze the great improvements that have taken place since that date, namely:

1. Scientific measure of human comfort.
2. Improved heat transmission surface for heating and cooling.
3. Improved methods of heat removal (refrigeration) including,
   a. New and improved types of refrigerating machines.
   b. New refrigerant media.

4. Improvements in the methods and apparatus for air distribution and humidity control.
5. Improvements and simplification in the method of dust removal (air filters).
6. Adequate lowering of objectionable sound level of ventilation and air conditioning systems (sound absorbers).
7. Development of reliable low cost unitary cooling and air conditioning equipment.
8. Extension of the application of air conditioning to new fields.

SCIENTIFIC MEASURE OF HUMAN COMFORT

An important scientific contribution to the field of air conditioning has been the determination of the effect of temperature, humidity and air motion upon human comfort and upon efficiency of industrial workers. The earliest work done in this field in this country was that of Dr. E. V. Hill, Chicago, Ill., who first published a synthetic chart in which all the component factors of ventilation were analyzed and evaluated. Among these was the relative effect of temperature and humidity. Dr. Hill had conducted a number of experiments in Chicago, which indicated that the wet-bulb temperature was a controlling factor in sensation of warmth. The validity of this finding, under certain conditions, was questioned by a committee of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, who were to pass upon the adoption of this chart. As a result, it was decided that the newly established Research Laboratory of the Society should determine the facts regarding this important phase of ventilation. A series of carefully conducted research studies were made, which have since become classical and accepted in all parts of the world as true measures of sensation of warmth, although the cooling power of the kata-thermometer, developed at an earlier period by Dr. Leonard Hill of England, is still preferred by some and is particularly useful in the study of ventilation conditions in deep mines. The results obtained by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS are of the greatest value in the practical application of air conditioning, as they provide a definite measure of the factors producing comfort in an air conditioning installation and permit definite standards to be established.

IMPROVED HEAT TRANSMISSION SURFACE FOR HEATING AND COOLING

The pioneer work in this field was probably done by L. C. Soule in conjunction with the author and other associates. The heat transmission surface shown in Fig. 2 was the first type of compact, highly efficient, light weight, non-corroding surface to be introduced and become commercially successful.

The development of a surface of this design had long been desired by air conditioning engineers and several attempts had been made to develop a surface of this type. The first essential was a non-ferrous metal such as copper, brass or aluminum, which would not be affected by the moisture in air conditioning systems. Reheater coils, when standing in conditioned air, corrode quickly, making it impossible to use wrought iron surfaces unless hot dipped after formation at excessive cost. Cast iron stood up well, but the rust forming on the surface was often objectionable by contaminating the air stream with iron

oxide dust. Where such conditions could not be tolerated, it had to be sherardized at considerable cost and only then with partly satisfactory results.

The weight and space occupied by wrought iron and cast iron heaters were also an important consideration, especially in the use of reheaters placed in the discharge area as was the preferred practice in certain air conditioning applications. If ferrous surfaces were employed for cooling coils, they had to be hot dipped to prevent corrosion and if direct expansion was to be employed, they had to be welded, largely on the job, because of the great weight and bulk on the larger installations of such construction. This was both expensive and unsatisfactory. A non-ferrous surface had to be provided with expensive materials, of which the cost, if prime surface alone was employed, would be prohibitive or else construction would be flimsy and therefore unsatisfactory. This meant the adoption of some form of extended surface in which a maximum amount of surface could be provided with a minimum amount of material and still retain the full thickness and strength of the parts subjected to mechanical and expansion stresses. The greatest obstacle was to provide a suitable assembly of the material for this purpose; which could easily be made hermetically tight and structurally strong, and at the same time allow freedom for expansion and contraction; that would endure high pressure and water hammer; that would also permit ease of assembly of smaller units to create a larger unit and which should be provided with standard casings to permit ease and accuracy of assembly in the field.

The development of a surface having these desirable characteristics was first accomplished with some success with the coil shown in Fig. 2, which was built in 1922 and presented to the public in 1924. There have been many improvements in the original construction, so that for the last 5 years or more this and other competitive products have met all these stringent requirements, providing a surface fully adaptable to the purposes of air conditioning at a far lower installed cost than the less satisfactory construction available in 1911. This development provided a low cost, highly dependable surface which made practicable the design of unit air conditioners and coolers. It should be considered one of the basic achievements in the air conditioning field since 1911. The complicated calculations, for determining the dehumidifying effect and heat transfer capacity of this surface used in cooling coils, have been standardized to give results of great accuracy. The method of calculation follows exactly the procedure given in the 1911 paper on Air Conditioning Apparatus.

IMPROVED METHODS OF HEAT REMOVAL

New and Improved Types of Refrigerating Machines

In 1911, and for several years thereafter, there were no practical methods of water cooling for air conditioning, except to direct the water over a series of expansion coils, known as Baudelot coils, into a tank. This required the use of an insulated space or Baudelot room, usually separate from the equipment. This arrangement was not only cumbersome, but expensive. However, at that time, it was the most satisfactory method of cooling water.

Another method which prevailed in some applications, where carbon dioxide was used, was to place coils in the spray chamber of the air washer with the addition of flooding nozzles. This design resulted in a combination of the old

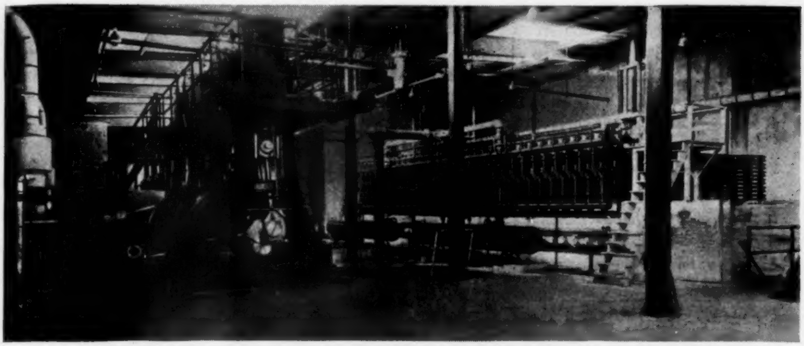

FIG. 3. 350 TON AMMONIA REFRIGERATION EQUIPMENT FOR AIR CONDITIONING (1918)

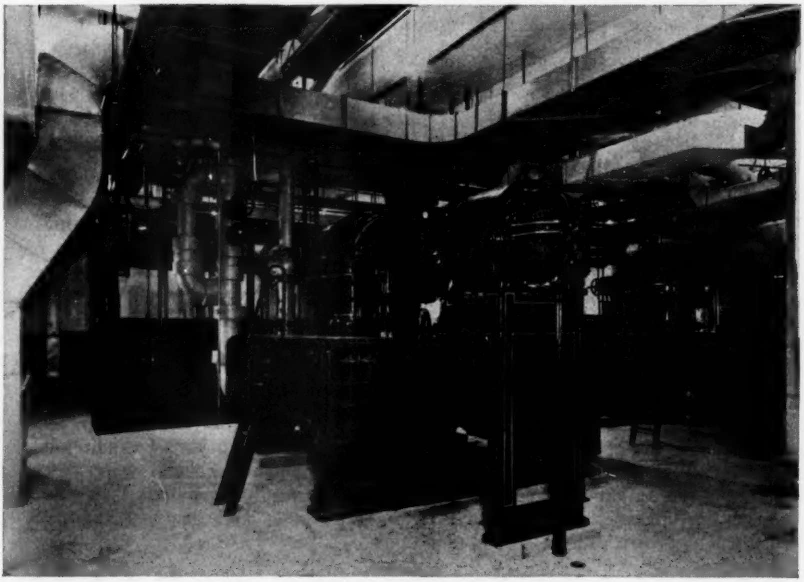

FIG. 4. 450 TON CENTRIFUGAL REFRIGERATION INSTALLATION FOR AIR CONDITIONING (1928)

bunker coil type of air cooling with the later type spray cooling. The success of this arrangement depended upon hand manipulation of expansion valves and was used only in comfort cooling where the variations in humidity were completely disregarded. The control of the load with this equipment presented difficulties as the air conditioning equipment was, of necessity, located at some distance from the refrigerating equipment. Hand operation of expansion valves was a practical obstacle which was unsatisfactorily met by giving a fixed setting day by day and running the refrigerating machine at full capacity. This method was wasteful of power and gave no control of humidity conditions. Automatic control was not used on this equipment and in fact was impractical. The operation with Baudelot coils, while permitting a perfect humidity control by the *dew point* method, was still open to the objection of hand manipulation of expansion valves and consequent waste in power consumption. Baudelot coils were generally allowed to ice-up at low loads and the suction pressure permitted to drop.

Air conditioning engineers had long realized the need of some improved type of refrigeration which would automatically respond, first to the demands made by varying loads in the air conditioning system, and second, would cool directly the water in the circuit to a definite point without danger of freezing and thus avoid the use of the expensive, cumbersome and otherwise objectionable Baudelot cooling room. Early types of refrigeration equipment used for air conditioning applications are illustrated in Figs. 3 and 4.

New Refrigerant Media

If a single factor were to be chosen as having the most outstanding influence in bringing about the wide spread adoption of air conditioning and summer cooling, existing today, it is probable that development of new and improved refrigerants would be given this honor.

Early in the art, engineers appreciated the hazards in the use of ammonia or sulphur dioxide to which most refrigerating machines were adapted, inasmuch as a break in the pipe would immediately convey asphyxiating gases to occupied spaces. For this reason the use of direct expansion coils was precluded and even with the use of Baudelot coils, these gases would be picked up by the water and disseminated into the air by the spray. Practically the only safe protection was to use an intervening medium such as brine with an intercooler to cool the water. This naturally was expensive to install, maintain and operate since it lowered the efficiency of the refrigerating cycle.

Carbon dioxide, which was generally acknowledged as a relatively safe refrigerant, presented great practical operating difficulties in an air conditioning system. The power consumption and quantity of condenser water were generally excessive, the automatic high operating pressures required heavy and expensive fittings, producing large refrigerant losses and rendered automatic control impracticable.

It will readily be seen from the foregoing that the state of the refrigeration art, 25 years ago, was most unsatisfactory from a standpoint of rapid promotion and widespread use of air conditioning for human comfort.

About the year 1902, LeBlanc in France and Parsons in England invented, and patented independently, a steam ejector system of refrigeration in which

water was made to boil at temperatures down to and below freezing by the use of brine instead of water.

About 1910 an attempt was made by the Westinghouse Co. to introduce

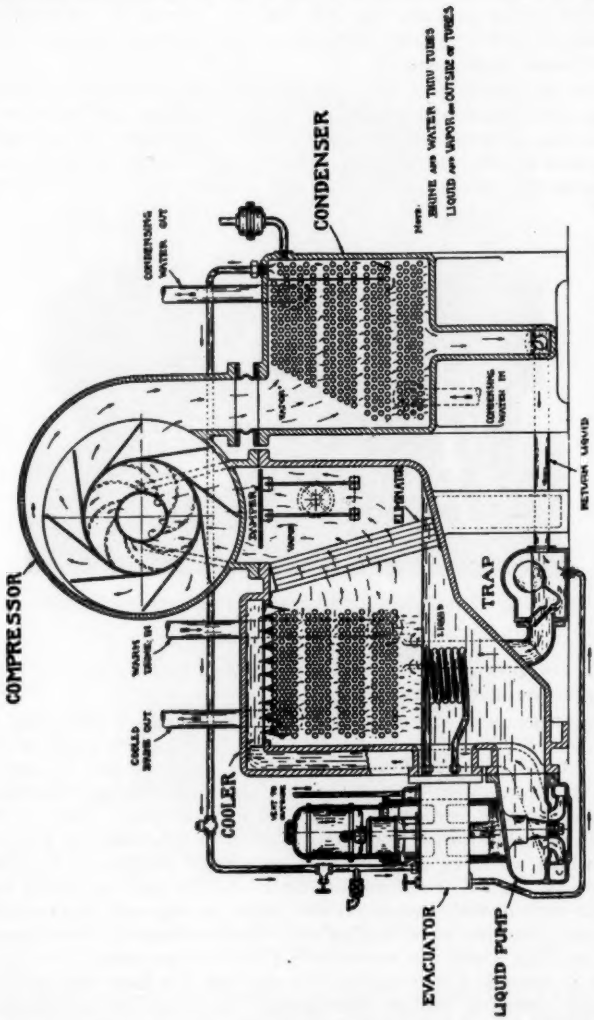

FIG. 5. TYPICAL DIAGRAM OF CENTRIFUGAL REFRIGERATING MACHINE (1922)

the LeBlanc process in this country. In 1912, the author and his associates investigated this system in an attempt to adapt it to air conditioning. Unfortunately, at that time, the system was not sufficiently perfected to make it eco-

nomical for high temperature refrigeration, that is, cooling the spray water direct, as the author had in mind. The cost was about double that of the corresponding ammonia system and the cost of operation was about in the same proportion, so that with regret, this possibility was abandoned. Air conditioning, at this period, probably had not sufficient volume to warrant necessary improvements in this system, which have since made it feasible for certain air conditioning applications.

In 1918 the author first saw the possibility of using low pressure nonhazardous refrigerants and a series of experiments were undertaken to develop a suitable type of compression system for such refrigerants. After considerable experimentation with other methods, a centrifugal machine (Fig. 5) was finally chosen as the best means for handling such refrigerants. It was first necessary

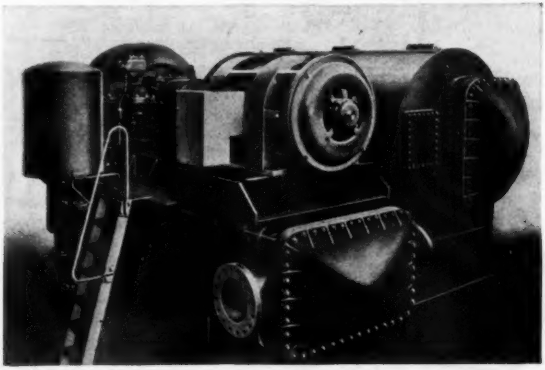

FIG. 6. 750 TON CENTRIFUGAL REFRIGERATING MACHINE FOR AIR CONDITIONING (1934)

to find a low pressure refrigerant of suitable characteristics in relation to hazard, pressure, corrosion, etc.

The construction of the centrifugal compressor was, at that time, well developed both in this country and abroad, and designs were available giving as high as 78 per cent mechanical efficiency. A greater use of this type of machine was made abroad than here. To adapt the centrifugal compressor to a refrigerant, however, involved many serious problems. Two of the most important were lubrication and the seal. Up to this time, no seal had been devised which would be effective on a centrifugal compressor handling a refrigerant gas. In 1921 a satisfactory design for the seal was finally completed and, with certain modifications, is the same as employed successfully today. Lubrication problems were studied and developed on the first experimental machine in 1922, which was successfully placed in operation in Newark in that year. It is interesting to note that this machine was later sold and is still in commercial operation for air conditioning. A large 750 ton centrifugal refrigerating machine for air conditioning applications is illustrated in Fig. 6.

This innovation in the refrigeration art opened up entirely new possibilities in the field of air conditioning, as it at once gave safety, provided for a varying

air conditioning load at substantially constant temperature, was much more compact than the previous system, permitting its introduction in many places where the older type of refrigeration could not be employed, and was easily made completely automatic. In other words, it seemed, at that time, to answer

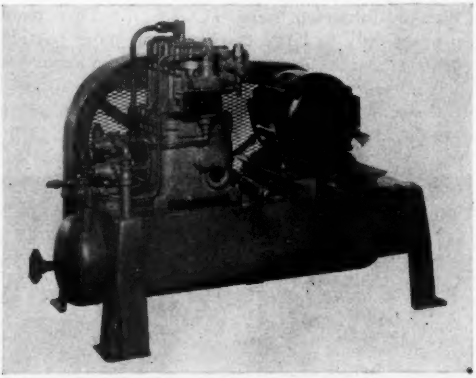

FIG. 7. DICHLORODIFLUOROMETHANE SMALL SELF-CONTAINED REFRIGERATING MACHINE (1931)

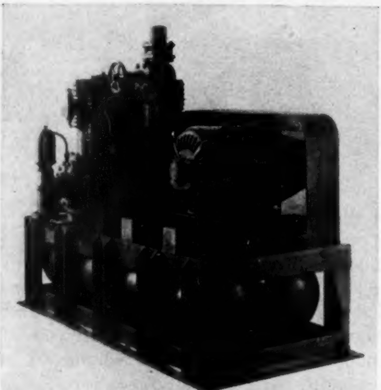

FIG. 8. 40 TON SELF-CONTAINED DICHLORODIFLUOROMETHANE REFRIGERATING MACHINE (1935)

completely the engineer's requirements for the ideal method of cooling for air conditioning. This was the first innovation employing a new refrigerant and system primarily suitable for air conditioning. During the subsequent period, however, there was a continued development using methyl chloride for small unitary automatic refrigeration and starting with the household unit. Some

methyl chloride machines were later employed in air conditioning, but few of these are now in use.

The next and possibly greatest stride was the development by Thomas Midgley, Jr., of a new series of refrigerants which were particularly adaptable to the reciprocating type of refrigeration machine. The best known of these refrigerants is dichlorodifluoromethane (CCl_2F_2). This resulted in a safe refrigerant for use not only in large machines but more particularly in a new range of reciprocating machines (Fig. 7) of smaller capacities, where there was a gap which had previously been entirely unfilled. This refrigerant was made available to the public in 1931. Not only did this refrigerant make application of reciprocating refrigerating machines (Figs. 8 and 9) to air conditioning safe, but in the smaller size machines it permitted a production and installation cost of practically half of that of ammonia machines of similar capacity. It also allowed the use of non-ferrous metals, of only moderate strength, in coolers and condensers. It enabled the use of direct expansion coils of the extended surface type directly in the air stream. Thus, at one stroke, was made practical a great extension of air conditioning to smaller installations, where, undoubtedly, the greatest volume of business is ultimately available.

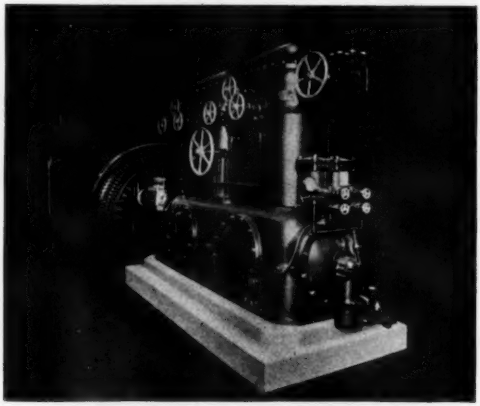

FIG. 9. LARGE SIZE DICHLORODIFLUOROMETHANE COMPRESSOR (1932)

In the last 5 years great improvements have been made in the mechanical adaptation of refrigerating machines to these new refrigerants. For smaller machines, mechanical seals are employed rather than stuffing boxes. Efficiencies have been greatly improved. Lubrication has been perfected and automatic controls are now functioning with great reliability. Automatic expansion valves are easily designed and constructed for the new refrigerant because of low working pressures. The suitable nonferrous metals have permitted a much more compact design, essential in unitary equipment.

Recently the steam ejector water cooler, as shown in Fig. 10, has also had a

successful development, resulting in a very simple and convenient arrangement for use where steam is available and where there is ample low cost water supply. While this system, in many cases, has disadvantages over the modern compression systems, it has been most satisfactory in marine work and on railroads. It is used extensively in conditioning of railroad coaches, due to the fact that a sufficient quantity of high pressure steam is always available and the condenser

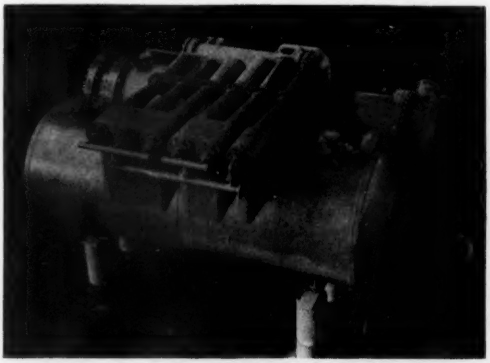

Fig. 10. Steam-jet Water-vapor Refrigerating Machine (1929)

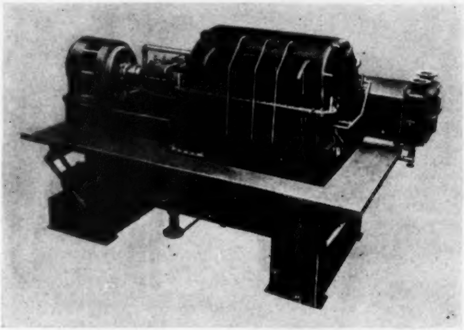

Fig. 11. Centrifugal Water-vapor Refrigerating Machine (1932)

problem is solved by the atmospheric condenser, operating on the evaporative principle. For railroad air conditioning each car requires its own power plant and therefore the steam ejector is the simplest type of compression that may be employed.

Since the invention of the LeBlanc process, there have been enormous improvements made in the efficiency of the steam ejector resulting in fair economy. There has also been a considerable simplification which has reduced the cost and

increased operativeness of the system. This is a particularly noticeable improvement in its application to railroad cars. It is only by such refinement that the equipment has been brought within the bounds of competitive practice.

Another interesting development is that of the water-vapor centrifugal compressor (Fig. 11), which is similar in principle to the LeBlanc process, with the exception that a centrifugal compressor takes the place of the steam ejector.

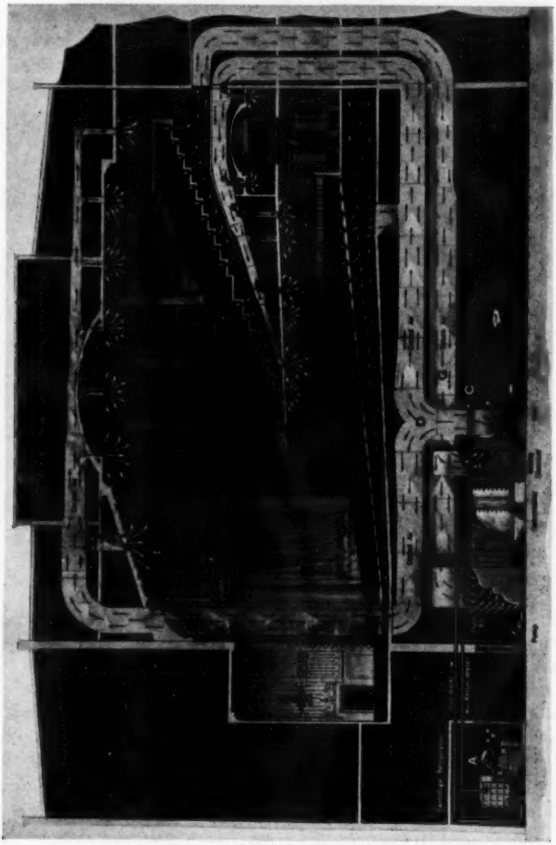

FIG. 12. DOWNWARD SYSTEM OF THEATER VENTILATION AND AIR CONDITIONING

This system requires smaller condenser surface and much less condensing water than the steam jet system because of the greatly reduced volume of vapor to be handled. Both of these systems have the practical advantage that the water is cooled directly by its own evaporation rather than requiring a heat interchanger, as in other systems of refrigeration. Naturally this effects a considerable economy in the thermal cycle. The principal difficulty to be sur-

mounted in the mechanical compression of water-vapor is the enormous volume to be handled, as well as the great centrifugal effect required, owing to the extremely low specific density of water-vapor as compared with other refrigerants employed in centrifugal equipment. However, from an engineering standpoint this is a most astounding achievement.

IMPROVEMENTS IN THE METHODS AND APPARATUS FOR AIR DISTRIBUTION AND HUMIDITY CONTROL

The former method of ventilation, for ordinary rooms, was to discharge the air through plain registers in the side walls, while for auditoriums the

FIG. 13. OVERHEAD TARGET AIR DIFFUSER

preferred construction was to discharge the air upward from a plenum chamber through mushroom ventilators underneath the seats. This was known as the *upward* system of auditorium ventilation. The first improvement in auditorium ventilation occurred about 1923 when the overhead (or downward) system first was successfully applied in a Los Angeles theater. The air was distributed by means of a series of outlets in the ceiling discharging vertically downward against ornamental plaques, as shown in Figs. 12 and 13. In this way the air was diffused horizontally and radially from each outlet forming a blanket of cooled conditioned air, which, by gravity, settled uniformly toward the floor, thus giving a uniform temperature from floor to ceiling throughout the auditorium. It is interesting to note that this type of outlet was first devised in 1913 to meet special ventilation requirements in a tobacco stemmery in Rich-

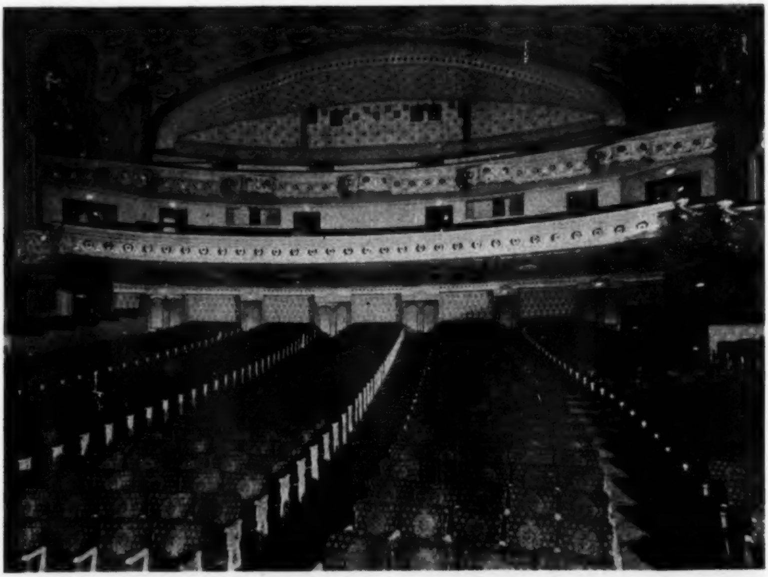

FIG. 14. EJECTOR SYSTEM OF AIR DISTRIBUTION

FIG. 15. SLOTTED OUTLET EJECTOR SYSTEM

mond, where a serious dust problem required large quantities of saturated air to be discharged, without drafts, into the room. This principle is most prevalent today for air conditioning and ventilation. It was a complete reversal of the older practice. In the Senate and House of Representatives at the Capitol building in Washington, D. C., this system now supplants the older system. The success of this method of distribution has been completely established by its general adoption.

Another type of ventilation which has been developed in the last 20 years, which has been most successfully applied to theaters and other buildings, is the ejector system of air distribution, as indicated in Fig. 14. This method of air distribution employs a high velocity jet, formed by relatively high pressure on a nozzle outlet. The air is distributed horizontally, high over the heads of the occupants of the room, and from the back of the auditorium toward the front. The effect of such a series of jets, discharging all in one direction, is to entrain a large volume of air, producing a secondary circulation, and a thorough mixture of highly conditioned air with recirculated room air. The secondary air circulation covers a large area and is uniformly discharged at extremely low velocity. This arrangement gives uniform temperatures within the theater and avoids objectionable drafts. The size of the nozzles may be modified to secure any length of blow required.

This system has many advantages, the principal one being that it permits use of relatively small quantities of highly conditioned air at low temperature which reduces duct sizes and cost of power for fan operation. It is interesting to note that this system was first devised, not for air conditioning of human comfort, but for improvement in the method of air circulation in the drying of materials. This principle was successfully applied by A. E. Stacey in the design of what is known to the trade as the *blue ribbon outlet*. This outlet is placed in the room in the form of a register and has horizontal or vertical slots which are formed in the shape of true rectangular nozzles, as shown in Fig. 15. These nozzles cause ample entrainment and permit the principle of the ejector outlet to be used in rooms of any size, giving greatly improved distribution and a low temperature differential between the air stream and the room. These effects are essential to avoid localized cold spots and objectionable drafts. The quantity of air may be regulated by closing off any desired portion of the discharge area.

Another application of this same principle is used in individual room cabinets in which the air is discharged upward toward the ceiling. These cabinets are usually placed under windows and are provided with an indirect, light weight heating surface in which steam or hot water can be controlled as in an ordinary radiator. The recirculated air induced by the ejector action is drawn through this heating element contained within the cabinet. When provided with small volumes of conditioned air at about ¼ in. pressure, the cabinet will ventilate, cool, heat and control the humidity of the room as required. The volume of air supplied to the room is also under manual control, so that it may be reduced any amount or closed off completely. This system has proven satisfactory in application to office buildings and provides an ideal system of air conditioning and heating.

Varying volumes of conditioned air in a large number of rooms connected to one system are provided for by the use of a static pressure regulator. This

maintains a uniform pressure in the main duct and branches, regardless of the amount of air delivered, so that the ejector action is maintained uniformly effective. This is an important development in exact control and satisfactory operation of air conditioning and ventilating systems.

Another important improvement made in the air distribution is a register which gives positive direction and distribution of air. The cold air, if introduced horizontally, is deflected upward at a slight angle toward the ceiling, assisting in the mixture and diffusion with room air. Still another form of diffuser is one which may be made to discharge the air at any vertical or horizontal angle and, at the same time, spread or diffuse it radially. This diffuser has many important applications where the exact distribution desired cannot be determined before installation.

Perhaps the greatest contribution to the art of air conditioning in the last 25 years has been the system of by-pass control devised by L. L. Lewis and improved by W. L. Fleisher. This process, in conjunction with artificial cooling, solves the problem of independent control of temperature and humidity of an enclosure. In some of the older attempts at cooling, humidity was ignored, and the temperature of the air introduced was varied by reduction of refrigeration effect upon the air itself. In such a system it is obvious that when small cooling effects are required there will be little reduction of air temperature with corresponding increase in relative humidity. In other words, when the temperature of the room was controlled by this method, the humidity would vary from possibly 50 per cent to a maximum of over 80 per cent under certain conditions of operation. It is obvious that with this method there was no real control over effective temperature. Moreover, abnormally humid air is objectionable from a comfort standpoint because it does not remove perspiration. The ideal effect in air conditioning is to prevent or remove sensible perspiration without too great a lowering of temperature. The desirable range of humidity for human comfort lies between 40 and 60 per cent and is, of course, related to the temperature as determined by the Comfort Chart. With automatic temperature control, a uniform as well as low relative humidity is an essential of true air conditioning for human comfort.

Previous to the advent of air conditioning for human comfort, it had been the practice in industrial installations to vary the volume of air introduced into the room in order to control the temperature without affecting the relative humidity. The relative humidity was determined by the relation of the dew point of the air to the room temperature. Another method often used supplementary to the above was to add heat artificially, that is, so as to maintain a constant heat load regardless of the requirements.

These methods could hardly be tolerated in air conditioning for human comfort. In the first place, in an auditorium a constant air volume is desired because any change in air volume causes a change in distribution which is objectionable. Second, supplementary heat is frequently not available and is undesirable because of unnecessary load imposed upon the refrigerating system at an increase in operating cost. It is permissible, from a ventilation standpoint, and also necessary for reasons of economy, that a considerable percentage of air be handled in the ventilation system. Advantage of this fact is taken, to by-pass a varying quantity of return air around the conditioner. By varying the amount of returned air by-passed, to the total amount passed through the

conditioner, it is possible to maintain a constant dew point at the conditioner and at the same time increase or decrease the amount of conditioned air without affecting the total volume of air circulated. This permitted varying the cooling effect according to the demand, and at the same time maintaining a constant dew point. A reasonably accurate control of relative humidity within the enclosure may be obtained by this method, which is now generally employed in all larger air conditioning installations.

IMPROVEMENT AND SIMPLIFICATION IN THE METHOD OF DUST REMOVAL

In addition to the air washing produced by the air conditioner, additional cleaning of air for ventilation purposes is quite generally provided by various

FIG. 16. UNIT AIR FILTER

types of air filters. These air filters operate on two distinct principles, first, by actual filtration through a porous substance such as specially prepared pulp paper which strains out and entrains the dirt, and second, by use of units or sections containing a filler of some character which is coated with oil or other viscous substance, providing a tortuous passage through which the air passes. The impingement of the dirt particles upon the viscous surfaces provides the necessary cleaning action required. The latter filters are of two types: first, the permanent type, as shown in Fig. 16, which requires servicing and cleaning, and second, the *throw-away* type, as shown in Fig. 17, constructed of especially cheap material, so that it may be discarded when it has become filled with dirt. There are also filters which are automatically cleaned which are of a more elaborate construction.

Modern air filters are effective in the removal of dust particles, but are inefficient in removal of smoke or fume. Their value and effectiveness were spectacularly demonstrated about a year ago, during the great dust storms

that prevailed in the West. Air conditioned buildings and railroad cars provided with dust filters went through these storms without any serious inconvenience, while other buildings and cars, not so provided, were almost overwhelmed with dust and subjected to great expense for renovation. The air filter has become a permanent part of a modern air conditioning system and will probably continue to be improved from a standpoint of effectiveness, installation and maintenance cost.

LOWERING OF OBJECTIONABLE SOUND LEVEL OF VENTILATION AND AIR CONDITIONING SYSTEMS

Until the advent of air conditioning for human comfort within the last few years, and particularly since application of air conditioning to sound studios

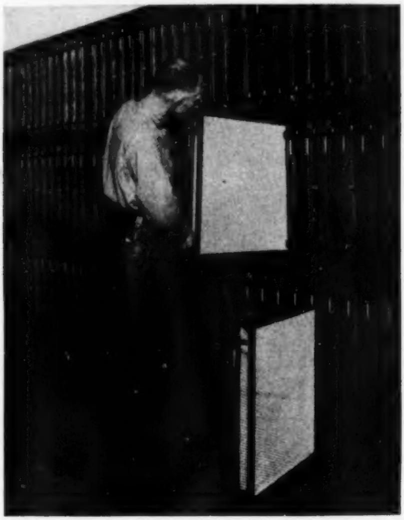

FIG. 17. THROW-AWAY TYPE AIR FILTER

and broadcasting stations, little attention had been directed to the subject of noise in the ventilating system. The noise problem was always obvious and frequently so objectionable that it was found impractical to operate the system after installation. This condition could not be tolerated in sound studios, which came with the talking movies, nor could it be tolerated in the broadcasting studios, which must be cooled and ventilated and *must* be quiet. This led to an early study, by air conditioning engineers, of sound prevention and absorption. Quieter ventilation equipment was demanded, and to a degree obtained, but principally the advance was made through a new technique of sound absorption, which has been made so perfect that the starting up or shutting down of the ventilating equipment cannot be detected within the quietest ventilated space.

The theory of sound absorption and the exact calculation of degree of sound absorption has been quite fully developed. Coefficients for various types of sound absorbers are quite accurately determined experimentally so that today the level of sound effects from a ventilating system can be calculated as exactly as the pressure drop in air ducts in the same system.

One of the most successful types of sound absorbers is of cellular construction made up of relatively small rectangular tubes of sound absorbing material nested in parallel to form a unit. These sound absorbers are interposed between the room openings and the ventilating equipment. Outstanding examples of an application of sound absorbers of this character are in the Capitol at Washington, D. C., and the National Broadcasting Studios at Radio City, New York.

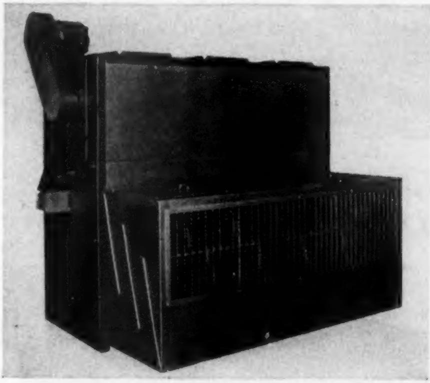

FIG. 18. AIR CONDITIONING UNIT FOR ALL YEAR SERVICE

The proper utilization of sound absorption methods makes possible application of air conditioning to homes and offices where otherwise air conditioning applications might only substitute one discomfort for another.

DEVELOPMENT OF UNITARY COOLING AND AIR CONDITIONING EQUIPMENT

The development of standardized unitary air conditioning equipment of various sizes, which can be completely assembled at the factory and easily installed, stands as a real achievement of the last 5 years of this quarter century. This has resulted from the availability of low cost, completely automatic refrigerating machines designed for the new and safe refrigerants, and also from the availability of compact, light weight heat transfer surface. These units are of various types, such as illustrated in Fig. 18, but are all designed to meet the need for a low cost installation which will require a minimum of supervision and servicing. They are especially adapted to use in small restaurants, stores, shops, isolated offices, etc. With the demand for such a unit provided there has, perhaps, opened up one of the largest markets for com-

mercial air conditioning. This market has been expanding with great rapidity within the last 3 years.

EXTENSION OF AIR CONDITIONING TO NEW FIELDS

The last quarter century has been particularly notable for the extension of the air conditioning industry into the field of cooling for comfort. The public

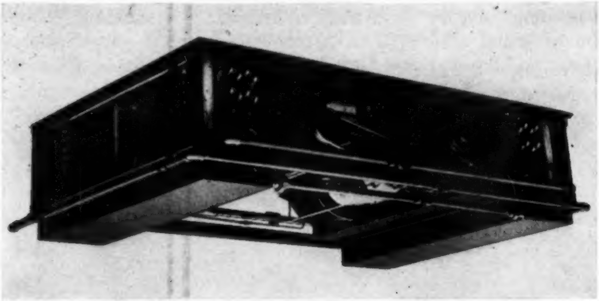

FIG. 19. AIR CONDITIONING UNIT FOR RAILWAY CAR

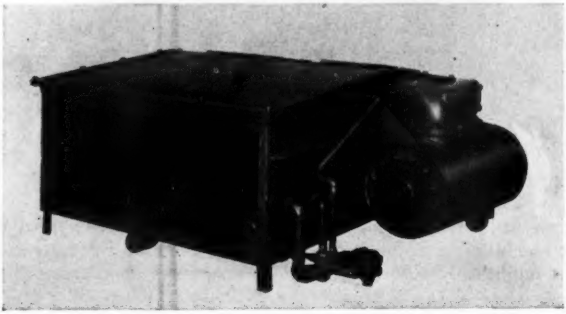

FIG. 20. STEAM-JET REFRIGERATION UNIT FOR RAILWAY CAR

first experienced the effects of air conditioning in the successful theater installations within the last 15 years, but perhaps the greatest aid to public acceptance has been the wholesale adoption of air conditioning by the railroads during the last 5 years. The rapidly growing and widespread use of air conditioning is being brought about by the advances in design and materials which make it possible to demonstrate to the small commercial user that air conditioning is a paying investment.

The first demonstration of the possibility of cooling a railway passenger car was made personally by the author at the Baltimore and Ohio shops in 1929. In 1930 two dining cars were equipped with successful air conditioning sys-

tems, one on the Baltimore and Ohio and the other on the Santa Fe. The following year three other installations were made. All of these early installations used ammonia, as no other satisfactory refrigerant was then available, and special precautions had to be taken against hazards in the use of this refrigerant. In the following year dichlorodifluoromethane became available and there was also developed a satisfactory system employing a steam ejector and water as the refrigerant. Both of these systems have received wide acceptance on nearly all the principal railroads throughout the United States and are shown illustrated in Figs. 19 and 20. There are a large number of cars being operated, with the steam ejector system of refrigeration for air

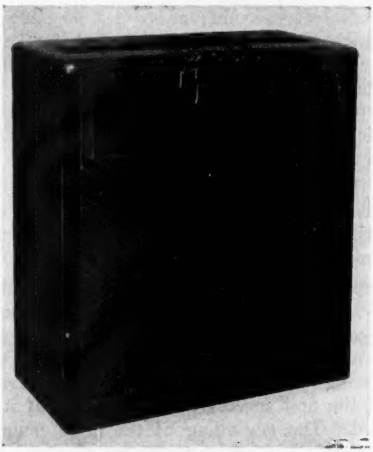

Fig. 21. Portable Self-contained Summer Air Conditioner

conditioning, on the South Manchurian Railway. As far as known, this is the only important installation anywhere outside of the United States and Mexico.

A recent notable extension of the art is the application to mines. A trial installation was made in the Morro Velho Gold Mines at St. John del Rey, Brazil, which was reported in the year 1922. By this application the working conditions in the mines were considerably improved and in 1929 this installation was augmented by an addition of centrifugal refrigerating machines and spray type air conditioners, located at the 6000 ft level. This latter installation has proven most effective, since it controls the temperature at the working level and permits the extension to any depth within the range of practical cost of mechanical operation.

More recently, in the Robinson Deep, at Johannesburg, South Africa, a 2000 ton installation was made for conditioning of the 400,000 cu ft of air per minute supplied to this mine. Later it is anticipated that booster installations will be made in the mine at the lower levels. The present depth of this mine is about 8500 ft and at this exploratory depth economic mining is impracticable without

air conditioning, owing to the excessive heat. With air conditioning, it is expected that the practical working depth of this mine can be increased many thousand feet. The Rand Gold Field, in which this mine is situated, at present yields over 50 per cent of the world's gold supply and an increase of the potential yield of this rich district by at least 50 per cent, through air conditioning, is of the greatest significance in maintaining a future stability in the world's economics. This application of air conditioning may eventually have greater influence on world advancement and prosperity than all other applications of air conditioning combined.

Air conditioning installations in homes and private offices are not yet numerous; but equipment is now provided which will satisfactorily meet the demands of this field. The most recent contribution is a portable air cooling unit, as shown in Fig. 21, which can be plugged into a light socket, without any other connections. This employs an air cooled condenser with a motor driven dichlorodifluoromethane compressor in the air conditioning cabinet. The greatest difficulties to surmount in this type of equipment are noise and improper ventilation. It is believed, in recent designs, that these obstacles have been largely overcome.

It is probable that the greatest strides in the general application of air conditioning have been made in the last half of the quarter century. Further improvements will probably be brought about largely by perfection of details of equipment, now already available.

And now in retrospect we see the progress of air conditioning development marked off in decades. 1901 may be said to mark the end of what might be termed The Azoic Age of air conditioning. That is the age in which there had been no development in the field either accomplished or begun. Ten years later, in 1911, air conditioning was acknowledged as a new art by the engineering profession. By 1921 the first successful modern type of installation for public comfort was designed. The beginning of the development of unitary equipment extending the benefits of air conditioning to small users came in 1931. Without thought of prophecy the year 1941 can be anticipated as marking the probable beginning of commercial success and public acceptance of complete air conditioning in the residential field.

In conclusion, appropriate recognition must be directed to the contributions to this development by all branches of the engineering profession. Continued interest and cooperation in development and refinement will accelerate engineering and commercial achievement with the resultant benefits to the public.

No. 1042

DEVELOPMENT OF TESTING APPARATUS FOR THERMOSTATS

By D. D. Wile * (*NON-MEMBER*), Detroit, Mich.

EVERY engineer is familiar with the important part that the thermostat plays in every day life. It forms the fundamental control feature of every air conditioning system and most ordinary heating or cooling plants.

Much has been done in recent years to improve the performance of thermostats and especially is this true of wall type thermostats used to control air temperature in a home or any other type of building. As in the case of most developments, it is important to have a suitable test method or *measuring stick* to determine the progress or improvement. This paper describes a novel test method which has been used to determine the accuracy of thermostats in the control of air temperatures, and to assist in the recent developments of new instruments.

For comparison the testing of an immersion type thermostat which has a remote bulb suitable for controlling the temperature of liquids may be considered. The bulb is placed in a liquid bath (Fig. 1) having an agitator and an accurate thermometer. Then by slowly changing the temperature of the liquid, the cut-in and cut-out points, or the so-called differential of the instrument may be determined.

The same procedure may be applied to the thermostat for controlling air temperatures (Fig. 2), and by exercising proper care, the cut-in and cut-out points may be measured accurately and thus the actual differential may be determined. If this thermostat is placed on the wall of a room a different condition is obtained. When the air is heating up, as shown in Fig. 3, the sensitive element lags behind, and the air temperature rises above the actual cut-out point of the instrument. The same condition of over-shooting exists at the cut-in point, with the result that the air temperature may fluctuate several degrees with a thermostat which operates in a liquid bath at a fraction of one degree.

The amount of this over-shooting of the air temperature depends upon air

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velocity and also on the rate of temperature change. The thermostat controls most accurately in fast moving air and slowly changing temperature.

In order to determine the accuracy of these instruments it is the usual prac-

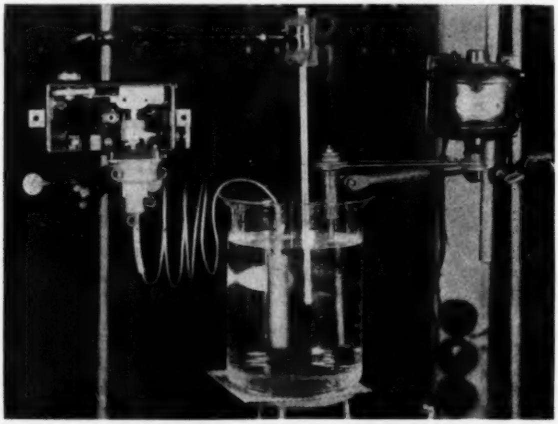

FIG. 1. IMMERSION TYPE THERMOSTAT TESTED IN LIQUID BATH

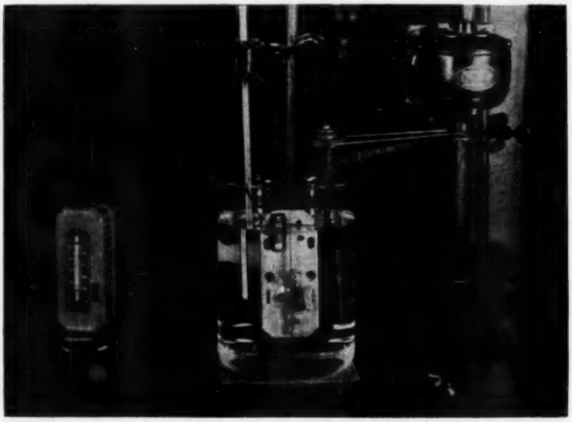

FIG. 2. WALL TYPE THERMOSTAT TESTED FOR ACTUAL DIFFERENTIAL IN LIQUID BATH

tice to operate in a specially constructed test room, but the results may often be unreliable and the test conditions may not even simulate practical operation.

Description of Apparatus

Several years ago it was decided to develop a reliable and simple method for

testing the response of wall type thermostats. One of the problems was to determine the conditions which surround the average wall thermostat both in respect to air velocity and rate of temperature change and then build equipment to duplicate these conditions in the most reliable manner. The apparatus shown in Fig. 4 is the result of this development. It consists of an insulated test chamber and a control panel and forms a completely automatic test apparatus.

Within the cylinder (Fig. 5) a special type fan forces a rapid circulation of air down the outside and up the inside of a cylindrical baffle tube. Within this tube is mounted another smaller cylinder whose ends are covered by a

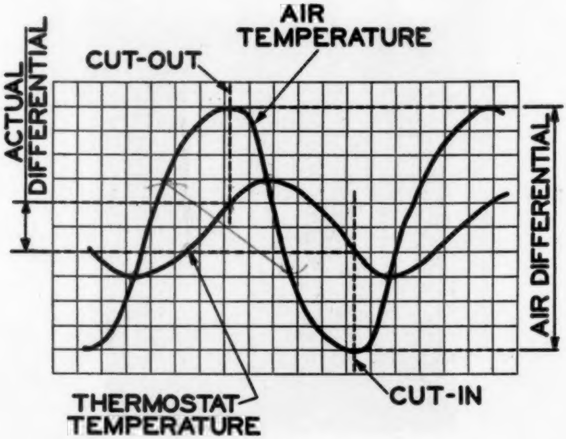

FIG. 3. LAG OF AIR TEMPERATURE

series of fine screens. These screens decrease the air velocity and insure a uniform velocity over the complete area of the tube.

The thermostat is mounted within this inner tube and may be subjected to a wide range of air velocities by regulating the number of screens over the ends of the tube and also by the speed of the fan. Air velocity over the thermostat is measured by an electrical anemometer, the construction of which will be described later.

Fig. 5 also shows the electric heater, the refrigerating coils for cooling and the resistance thermometer. This thermometer consists of a long length of nickel wire which changes its electrical resistance with any change in air temperature. As shown in Fig. 6, the wire is wound into a long spiral and mounted around the inside wall of the baffle tube. This construction provides a large amount of wire surface to come in contact with the air and permits the thermometer to follow the air temperature even though it be changing at a rapid rate. This is an important consideration since any discrepancy in the measuring instrument reflects directly in the test results.

Before describing the control circuit, it is desirable to explain the general

operation of the system. When the thermostat cuts in and out it does not turn the heat on and off in the usual manner as this arrangement was not sufficiently reliable. Instead it was decided to make the temperature increase or decrease at a definite rate, as shown in Fig. 7. When the thermostat calls for heat, the temperature starts to increase at a uniform rate and continues until the cut-out point is reached and then starts decreasing at the same rate.

Obviously this condition is not the same as exists in practice since no such uniformity is ever found in practical operation. However, this is a condition that can be accurately controlled and reliably reproduced at any time. It thus provides an excellent basis for comparing one type of design against another

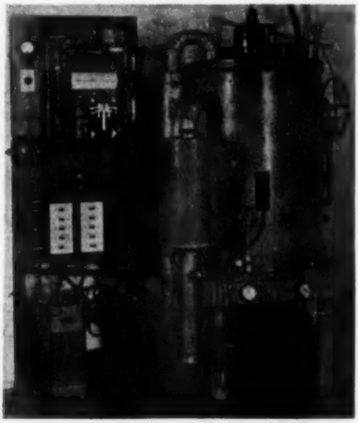

FIG. 4. NEW TEST APPARATUS AND CONTROL PANEL

and furnishes the necessary measuring stick for developmental work. In order to prove the reliability of the apparatus, tests have been repeatedly conducted on the same thermostat at intervals of more than one year apart and, with the same test conditions, obtaining identical results. Moreover, experience has shown that by using comparable air velocities and observed rates of temperature change, the test results predict the performance of the thermostat in the field.

The performance shown in Fig. 7 is accomplished by an electrical control circuit comprising the resistance thermometer shown in Fig. 6, a sensitive galvanometer and a slide wire mechanism connected together in the form of the Wheatstone Bridge. The apparatus is mounted on a control panel as shown in Fig. 8. At the top is the slide wire and below this is a recorder drum, while at the bottom is the galvanometer. These units are driven through a system of gears by a synchronous motor shown at the lower left hand corner.

The contact on slide wire (Fig. 9) moves along at a constant rate of speed and sets the pace for the rate of temperature change in the cabinet. This unit is driven from the main vertical drive shaft through change gears, which pro-

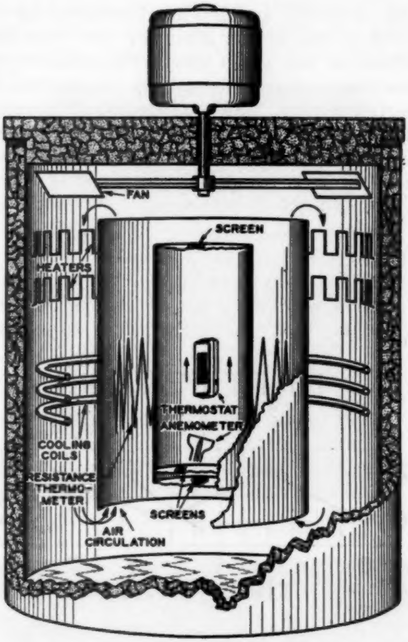

FIG. 5. CROSS-SECTION OF TEST CABINET

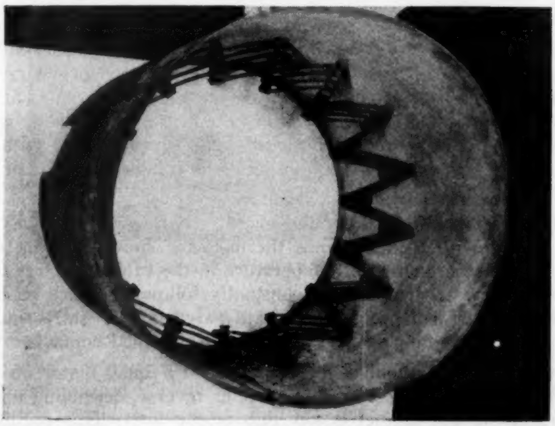

FIG. 6. VIEW OF RESISTANCE THERMOMETER

vide any desired rate of motion, and thus any corresponding rate of temperature change. The position of the contact along the slide wire determines the temperature in the cabinet at any given time. This slide wire is very accurate, requiring a complete turn to change the cabinet temperature one degree. There are 35 turns in the slide wire with a corresponding temperature range of 35 F; therefore, a manual adjustment is provided to change the operating range when running tests at very high or very low temperatures. This adjustment is located on the right of the mechanical slide wire, and is seldom changed except when special tests are required at very low or very high temperatures.

The unit in the upper left hand corner is a solenoid which is energized when the thermostat in the test case calls for heat and is used to reverse the rotation of the upper portion of the main drive shaft. This view shows the solenoid in the de-energized position with the small bevel gear in mesh and the upper shaft driven in reverse direction. When energized, the solenoid raises the upper

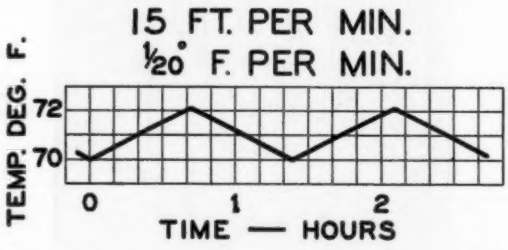

FIG. 7. REGULATION OF AIR TEMPERATURE IN TEST CABINET

part of the shaft, lifting the upper bevel gear out of mesh and allowing a positive clutch to engage, thus driving in the opposite direction than when the bevel gears are in mesh.

The recorder drum (Fig. 9) is rotated by gears driven from the main drive shaft. The recording stylus, however, is driven from a secondary shaft connected to the slide wire mechanism. The drum therefore rotates at a uniform timed speed, while the marking pen follows the motion of the slide wire contact and makes a record of the time of thermostat operation along with the actual variation of air temperature, thus obtaining a complete automatic record of thermostat performance.

The galvanometer (Fig. 10) is the nerve center of the system. It is connected in circuit with the resistance thermometer shown in Fig. 6 and may instantly respond to a change in temperature in the cabinet of less than 0.02 F and insures that the air temperature constantly follows the pace set by the slide wire. When out of balance, this delicate galvanometer starts into action a positive mechanical motion to operate a set of electrical contacts.

These contacts in turn control the operation of a small motor on the voltage regulator (Fig. 11) causing it to start, stop, or reverse, depending upon whether the temperature in the cabinet is too high or too low. This regulator, therefore, supplies current to the main heater in the cabinet at exactly the required

amount to offset heat leakage through the cabinet walls and also to counterbalance the refrigeration load. The refrigerator operates continuously throughout both heating and cooling cycles and makes the apparatus independ-

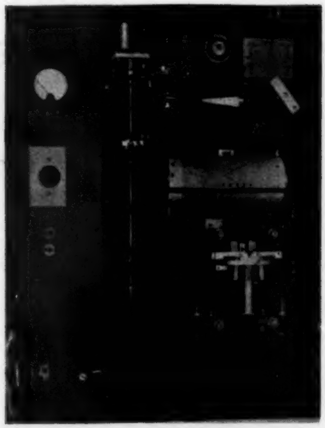

FIG. 8. VIEW OF CONTROL PANEL

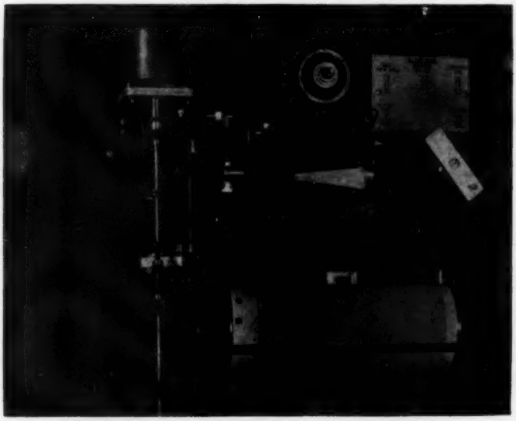

FIG. 9. SLIDE WIRE AND RECORDER DRUM

ent of outside temperatures. The insulated walls permit operation at relatively low temperatures.

Fig. 12 shows a schematic wiring diagram, the complete diagram being somewhat more complicated. Relays are used so that only a small current need be carried by the thermostat contacts and the relays further provide for testing two or three wire instruments or refrigeration type instruments where the con-

tact action is reversed. Any current desired may be thrown across the thermostat contacts and it may be made to operate under load conditions.

Testing Method

During test when the thermostat cuts in, the temperature must instantly

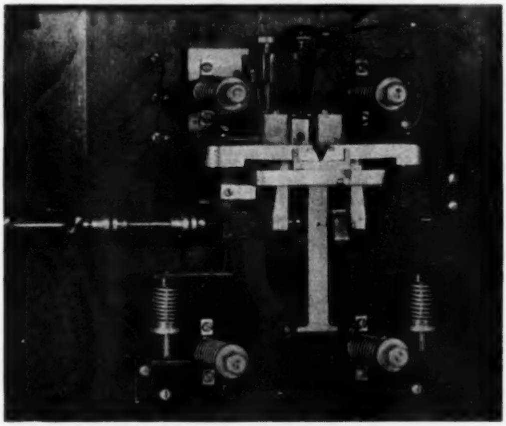

FIG. 10. POWER DRIVEN GALVANOMETER

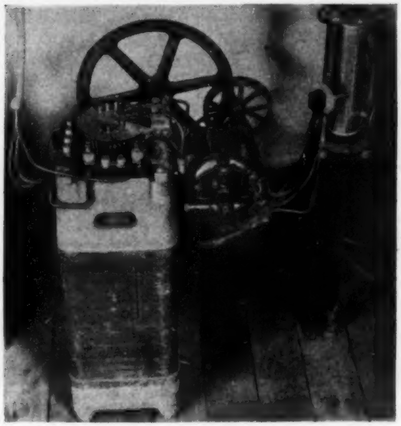

FIG. 11. MOTOR DRIVEN VOLTAGE REGULATOR

change from a decreasing to an increasing rate and this involves a considerable change in the heat input to overcome the thermal capacity of the air and other parts within the cabinet. While the temperature is falling, the cabinet parts are giving up heat and while rising they must receive heat. This instantaneous

demand is met by a secondary heater operated by the thermostat through a relay. The heater turns on when the thermostat calls for heat and supplies the correct amount to satisfy the sensible heat load of the cabinet parts. Once properly adjusted for a given rate of temperature change, the heater needs

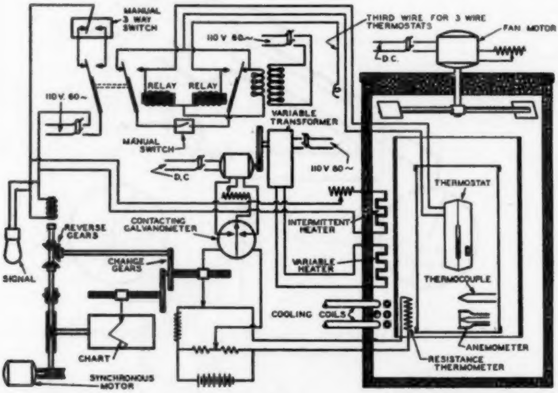

FIG. 12. SCHEMATIC WIRING DIAGRAM OF APPARATUS

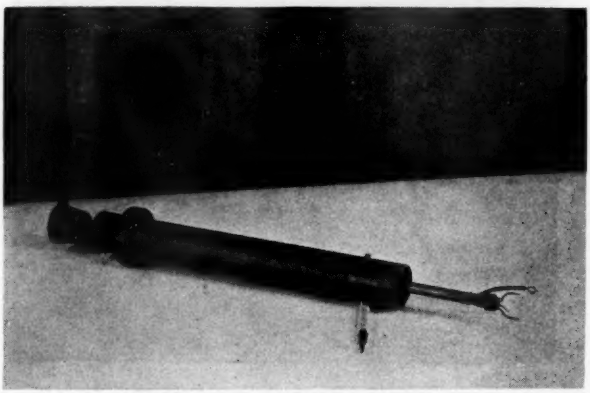

FIG. 13. ANEMOMETER USED IN TEST CABINET

no further attention since any small variations are corrected by the main heater.

This arrangement may appear to be a complicated layout, but actually it works out in practice simply. After the equipment is set into operation, it continues automatically without any attention. In addition, automatic safety devices are incorporated which not only protect the equipment, but also shut

it off after a predetermined length of time. Often tests are started near the end of the day and allowed to run automatically to completion. At the end of the test the equipment stops operating and is ready the next morning with a chart showing the performance of the thermostat under test.

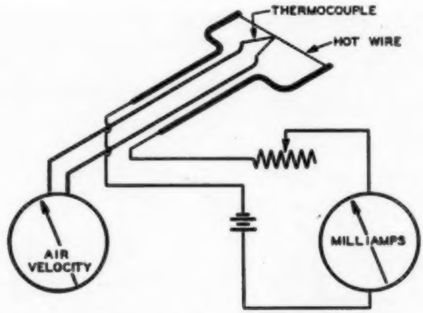

FIG. 14. WIRING DIAGRAM OF ANEMOMETER SHOWN IN FIG. 13

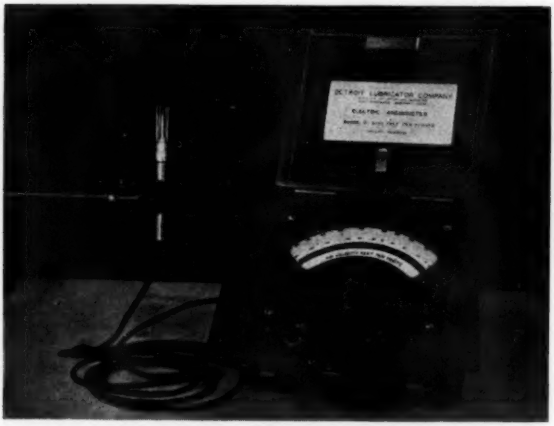

FIG. 15. PORTABLE ANEMOMETER

Electrical Anemometer

The anemometer for measuring the air velocity presented a problem since it was necessary for it to give accurate results at extremely low velocities and also it had to be of a construction which would not interfere with the smooth flow of air near the thermostat. The hot wire type of electric anemometer was selected to meet these requirements and it was necessary to conduct an elaborate investigation in order to develop this type of instrument.

Fig. 13 shows the type of anemometer used in the test apparatus. This device is arranged in a protecting tube so that it may be withdrawn for protection against damage during the time when a thermostat is being installed in the case and then merely pushed into position when the test is started.

A schematic diagram of this anemometer is shown in Fig. 14. It consists of

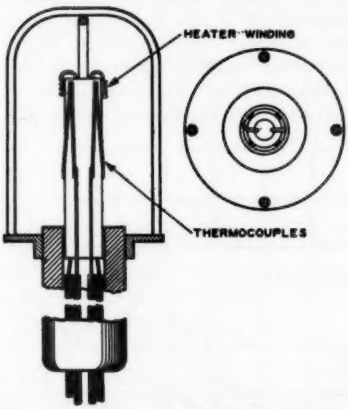

FIG. 16. DETAIL OF MEASURING TIP OF PORTABLE ANEMOMETER

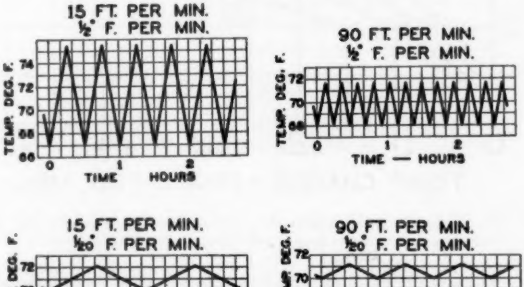

FIG. 17. RESULTS WHEN INSTRUMENT IS TESTED AT VARIOUS AIR VELOCITIES AND RATES OF TEMPERATURE CHANGE

a small resistance wire known as the *hot wire* stretched between two supports and supplied with electric current which may be accurately measured by a milliammeter. Near the center of this hot wire is attached a thermocouple made of extremely fine wire and connected to a very sensitive potentiometer which is capable of measuring one-tenth of a millionth of a volt. The potentiometer reading indicates the temperature of the hot wire to an accuracy of 1/200 F. This hot wire, being of small diameter, is easily cooled even by a

slow movement of air, and therefore the reading of the potentiometer may readily be calibrated to show the actual air velocity passing over the hot wire.

Portable Anemometer

The anemometer shown in Fig. 13 was satisfactory for use in the test cabinet but it was not suitable for making measurements in the field, and it was therefore necessary to determine the actual air velocity existing in normal installations in order to establish proper test conditions. A portable anemometer was subsequently developed which could be depended upon to give sufficient accuracy in actual installations and yet permit speed and ease in taking readings. Fig. 15 shows the portable type anemometer as it was finally developed. It consists of a measuring tip connected by means of a flexible cable to a meter box calibrated in actual air velocity. The complete equipment including the batteries, switches, rheostats, etc., is arranged in the meter box.

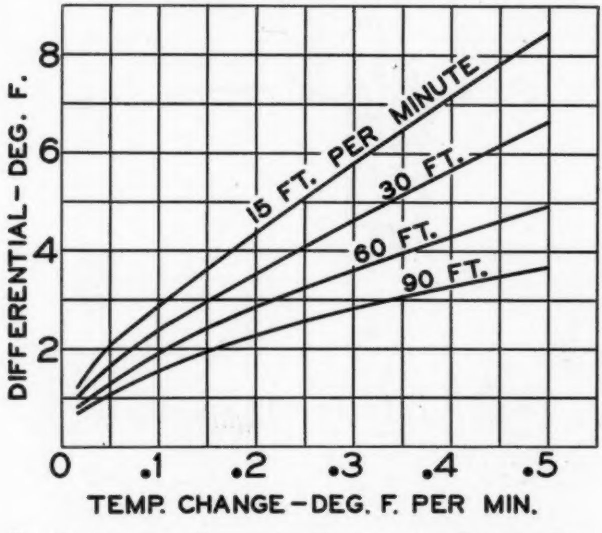

FIG. 18. EFFECT OF VARIOUS AIR VELOCITIES AND RATES OF TEMPERATURE CHANGE ON A WALL TYPE THERMOSTAT

Although the portable instrument works on a hot wire principle, the construction of the measuring tip (Fig. 16) was modified considerably in order to obtain sufficient accuracy with a portable meter. The hot wire used in this instrument is wound around a small tip and over the top of several thermocouple junctions. These thermocouple junctions being connected in series permit the production of an appreciable voltage with a small amount of heat generated by the hot wire. Some idea of the size of this tip may be realized by noting that the largest outside diameter is less than the size of a fountain pen.

By the use of this tip construction, the actual temperature of the hot wire rises only a few degrees above room temperature and therefore does not in

itself disturb the air motion. This anemometer proved entirely satisfactory for measuring the comparatively slow air velocities encountered near the walls of residences and other buildings. It has also been used in various types of laboratory tests and at one time was used in a large mine to make a complete survey of air velocities existing in the various tunnels and working spaces where the velocities were too low to be measured by the usual Pitot tube.

In houses heated by radiators or gravity circulated warm air, the velocities at the walls where thermostats are usually located were found to vary from nearly 0 to 40 fpm. The average for good locations was between 15 and 30 fpm. Velocities vary with the heating load, being much more rapid during extremely cold weather when the heating plant is operating under heavy load conditions.

Application of Test Apparatus

The effect of air velocity and rate of temperature change is shown in Fig. 17 by the four tests made in the apparatus at widely varying conditions. The same

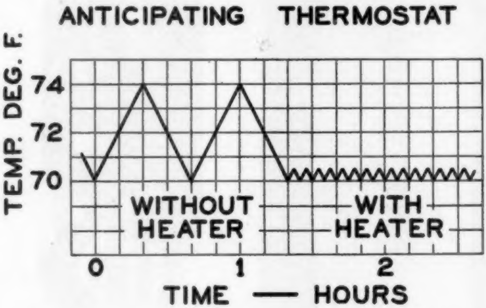

FIG. 19. EFFECT OF COMPENSATING HEATER ELEMENT ON THERMOSTAT OPERATION

thermostat was used in each of these conditions without any change of adjustment. It was carefully checked before and after the tests to insure that it had not changed. In Fig. 17 the first two or three cycles have been omitted from each of the curves, as these first cycles are usually irregular while the thermostat is settling down, after which the cycles are regular regardless of how long the test is continued. It is generally possible to complete a run in 3 hours. The results of a complete series of tests are shown in Fig. 18 and demonstrate the effect of air velocity and rate of temperature change on thermostat operation.

The test apparatus has recently been of value in a long series of tests on compensating or heat accelerated type thermostats. A few of these test results will be shown because they demonstrate the value of proper test equipment in this type of development work.

The compensating or heat accelerated type of thermostat has a small heater element inside of the case, which applies heat to the sensitive element or blade as soon as the instrument cuts in. The heat supplied to the sensitive element anticipates the warming up of the room air and may be adjusted so that the

thermostat operates on a small differential. This action is shown in Fig. 19, where it will be noted that when operating without the heater the instrument had a differential in air of 4 deg but this reduced to ½ deg when the heater was thrown into the circuit. This type of instrument provides a positive action while maintaining close control of the air temperature. The action of the heater generally makes it unnecessary to locate the instrument in a region of high velocity and in some installations this has certain advantages since the most desirable thermostat location from a standpoint of air movement is in some cases the most undesirable location from the standpoint of appearance.

In order to satisfy various types of installations, the amount of heat imparted to the blade must be variable and there are several means of accomplishing this result. Fig. 20 shows the construction of a thermostat with the compensating

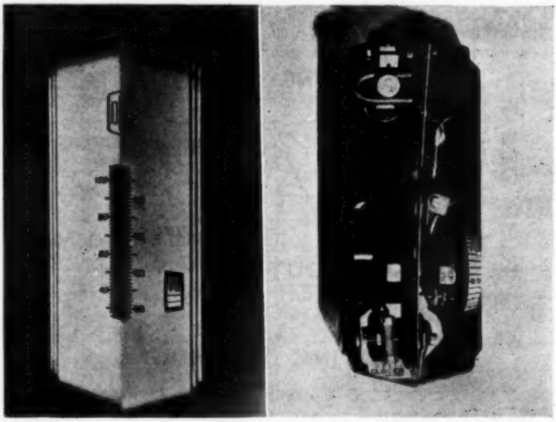

FIG. 20. VIEW OF A COMPENSATING TYPE THERMOSTAT

heater element on an adjustable mounting so its distance from the sensitive blade may be varied. In this instrument no adjustment is provided for the voltage to the heater element and it is purposely kept constant by means of a third wire brought in from the transformer. The heater element is separated from the control circuit and therefore the compensating effect is independent of the electrical load on the thermostat. The compensating effect is regulated by changing the position of the heater element relative to the sensitive blade and thus provides a means of obtaining the desired operation on any type of installation.

If the heat imparted by the heater element to the blade is increased sufficiently, the thermostat will cut in and out at intervals without requiring any change in the air temperature. Under some conditions of operation this is desirable, especially where the heating plant has a long *carry-over*. The thermostat may be made to compensate entirely for the *lag* and *carry-over* in the heating plant and provide uniform temperature where such accuracy of control is desired.

The design of this instrument involved many test runs in order to determine the proper amount of watts supplied to the heater element and also the effect of various positions of the element. The size of louvers or vent holes in the outside case also plays an important part. This information was easily obtained by the use of the test apparatus described in this paper. Fig. 21 shows a sample of the test results and indicates the effect on thermostat operation of both the position of the compensating heater and the watts supplied to it.

These data were necessary in the design of the thermostat and it would have been a most difficult, if not impossible, task with the usual method of test.

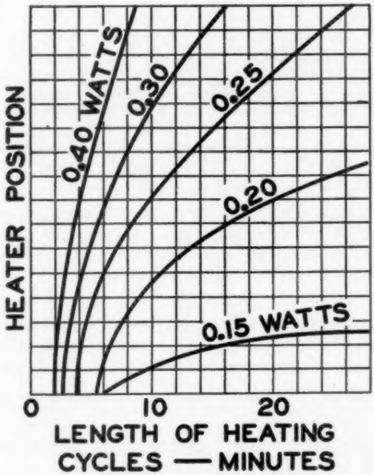

FIG. 21. RESULTS AT VARIOUS HEATER LOADS AND HEATER POSITIONS

The special test apparatus made it possible to obtain these data in comparatively short time. This apparatus has been in almost constant laboratory use for a period of several years and has contributed an important part in the continued improvement of thermostat design. It has reduced the testing of thermostat response to an exact science and provided the necessary measuring stick.

DISCUSSION

B. E. SHAW (WRITTEN): An exceptionally fine article describing the development of a thermostat test device has been presented by the author of this paper. Such developments are typical of the strides Research has taken in modern manufacturing by companies possessing the willingness and the wherewithal in money and man power to pursue such methods. As a result engineering, science and more important still the comfort of mankind takes a forward stride.

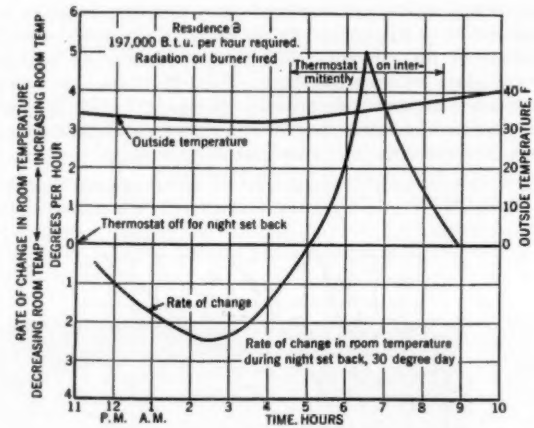

FIG. A. RATE OF CHANGE IN ROOM TEMPERATURE DURING NIGHT SET BACK, 30 DEGREE DAY

Commenting specifically on the design of the test apparatus it would appear that the equipment is rather complicated and would require a good deal of manual manipulation every time a thermostat or change of condition is investigated.

Would Mr. Wile tell us how much effect on anemometer velocity determinations

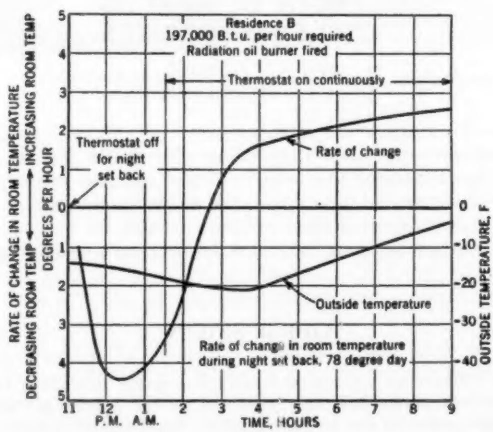

FIG. B. RATE OF CHANGE IN ROOM TEMPERATURE DURING NIGHT SET BACK, 78 DEGREE DAY

changing ambient temperature had; and also how was the electrical anemometer calibrated?

Commenting upon Fig. 18 it is questioned whether the data presented therein are of any value from the design standpoint. The only portion of the curve that can be used extends to the left of (or is less than) 0.1 F temperature change per minute. For instance on one installation investigated the rate of temperature decrease after night set-back varied from 0 to 4.5 F per hour (0.075 F per minute) and 0 to 2.5 F increase per hour (0.042 F per minute) after a burner operation and morning pick-up—all this at an average outside temperature of − 1 F. With an outside temperature of 35 F the data recorded would indicate a maximum rate of temperature decrease to be − 2.5 F per hour (0.042 F per minute) and a maximum rate of temperature rise to be 5 F per hour (0.083 F per minute) as shown in Figs. A and B.

It is apparent that the rate of temperature change is dependent upon such factors as outside temperatures, burner operating time, whether the installation is heating up or cooling down, and finally on thermostat differential. Hence, it is questioned whether or not we can say (as Fig. 18 indicates) that the differential is a function of the temperature rise, but rather that the temperature rise is a function of the set differential of the thermostat.

L. P. Hynes (Written): This paper should be of great value to engineers interested in laboratory tests of thermostats. It should also be studied by every engineer who has anything to do with specifying or installing thermostats because it clearly indicates the variable conditions which affect actual thermostatic performance. Too often the actual operating characteristics of a thermostat are ignored in blissful confidence that any thermostat will accomplish the desired temperature regulation if installed at the right location and set for the desired temperature.

Heating engineers are constantly dealing with questions of heat transfer from one medium to another, as for instance from metal to air, or from air to metal. They take into consideration radiation, conduction, convection, and the specific heat of materials. In other words, they reduce their heating calculations to basic principles and carefully compute the heat energy required to effect a desired result. However, they often forget these basic principles when considering temperature control and overlook the obvious fact that a considerable amount of energy is required to operate a reliable contact mechanism of the voltage and ampere capacity required for a modern thermostat. They also forget how little energy is represented by a temperature change of 1 F in a cubic foot of air.

The problem is not so serious with thermostats designed for immersion in bodies of liquid because liquids have relatively large thermal storage capacity and offer good contact with and thermal conductivity to the sensitive element of the thermostat. Consequently, in any tank or reservoir the thermostat can usually follow closely the actual temperature changes of the liquid.

More difficult problems occur with thermostats installed for the control of air temperatures, and this paper gives much useful information on this subject. The laboratory apparatus described is admirably adapted to accurate and reliable determination of the exact characteristic of any particular thermostat. It is, of course, not intended nor adapted for commercial testing of thermostats in production manufacture. It is, however, an accurate and reliable standard for checking and calibrating purposes. All the principal variables are taken into consideration and independently controlled so that any combination of conditions can readily be set up and duplicated whenever desired.

It is often possible to adjust thermostats for operation in air by immersing them in a bath, provided actual operating conditions are first checked and a proper differential for operation in the bath is determined upon. For the testing of thermostats intended for controlling air temperatures, a simple and practical method is to use a cabinet or duct with a recirculating air system and mixing dampers to gradually

admit warmer or cooler air. By controlling the rate of temperature change and the velocity of air circulation, any desired operating characteristic can be approximated. When assembling such equipment it is necessary to have some satisfactory way of checking all variable factors. The method described by the author of this paper should do this accurately.

The electric resistance thermometer method of controlling the temperature and the hot wire type of anemometer are very skillfully used by Mr. Wile for controlling and measuring purposes. The use of the power driven galvanometer relay for voltage control of the electric heaters, through a slide wire resistance, is an excellent and reliable way to vary the heat input.

The portable anemometer described is an ingenious instrument that should prove useful in determining air velocity which is such an important factor in thermostat performance. The extremely small size of the sensitive measuring tip makes it possible to measure air velocities in spaces too small for ordinary methods. The data given regarding the velocities of gravity circulated air along walls is interesting and useful. An important part of this paper is the stressing of the effect of air velocity upon the operation of a thermostat and the temperature lag between the actual inherent instrument differential and the air differential. In specifying and using thermostats we all need to keep these facts in mind and it is helpful to understand clearly the basic principles back of this problem of lag.

The factors here are simply the heat storage capacity of the thermostat and its mountings due to mass and specific heat in comparison with the specific heat, velocity and rate of temperature change of the air moving over the thermostat. A theoretically ideal thermostat would be one having no mass because mass means heat storage capacity and metal has a high heat storage capacity compared to air. In addition to the actual heat storage capacity of the mass of the sensitive element of the thermostat, the effect of the mounting for the element, the protective casing, and of the adjacent wall must be overcome. Conductivity and radiation play their part and a relatively large amount of air must flow over the thermostat to produce the heat transfer necessary for prompt operation.

The lag between the actual differential of a thermostat as measured in a bath and an air differential, as shown in Fig. 3, is of especial value. The charts in Figs. 17 and 18 are particularly interesting because they show clearly the effect of air velocity and rate of temperature change upon the operating differential of a thermostat.

An excellent description is also given of the compensating or heat accelerated type of thermostat with a built-in electric heater. The heater element adds one more variable factor which makes possible many new combinations and permits numerous unique operating results. This type of instrument is often useful for securing some particular desired cycle of operation but the correct application of the compensating principle requires careful analysis of the application under consideration.

In spite of the tremendous growth of automatic control I believe we are destined to see still greater developments and one of the essentials to such advancement is a thorough understanding of all the variable factors which enter into the problem. Another essential is accurate means for measuring and checking the variables independently and in various combinations. The laboratory methods described in this article do this in a thoroughly scientific manner and I believe this paper is a real contribution to the very limited literature on the subject of thermostatic performance.

No. 1043

PYRHELIOMETERS AND THE MEASUREMENT OF TOTAL SOLAR RADIATION

By L. A. HARDING * (*MEMBER*), BUFFALO, N. Y.

PHYSICISTS and engineers have for the past 100 years been interested in schemes for the measurement of total solar radiation per unit area received on the earth's surface. Interest in this matter is obviously natural, as life in its various forms existing on this planet is dependent on the cyclic quantity of solar energy received.

The form of curve obtained on a clear sky day by plotting the intensity of solar radiation with time base is well illustrated on page 277, A.S.H.V.E. TRANSACTIONS for 1932, also page 151, A.S.H.V.E. TRANSACTIONS, 1930.

Various engineers have also, from time to time, interested themselves in experimental plants for the direct or indirect utilization of these rays for water heating and power purposes. Among these may be mentioned the names of Mouchot (1878), W. Adams (1876), M. A. Pifre (1880), John Ericsson (1883), J. Harding (1883), Eneas (1901), Willsie (1904), Shuman-Boys (1913), and recently Georges Claude (1930), whose large scale experiment in Cuba is well known in this country.

Domestic water heating by solar radiation is practiced on a considerable scale in this and other countries and engineers interested in the subject of air conditioning have also quite recently concerned themselves with this matter.

The so-called *sun effect* with comfort cooling installations constitutes, at times, a considerable portion of the heat to be extracted in order to maintain a desired constant temperature condition within the structure. It is now generally recognized as one of the items which must be taken into account by the designing engineer of comfort cooling installations.

The *Smithsonian Institution* maintains a number of stations in various parts of the world where continuous observations of solar radiation are made, while some of the U. S. Weather Bureau stations are now supplied with pyrheliometers and it is assumed that in the future all such stations will be so equipped.

The term *plate,* as hereinafter used, is assumed to be the heat absorbing element exposed to the sun's rays. This element may be a solid of any shape; however, an inverted cone possesses certain advantages as it practically eliminates the question of the efficiency of the absorbing surface as such.

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* Pres., L. A. Harding Construction Corp.

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, June, 1936, Buck Hill Falls, Pa.

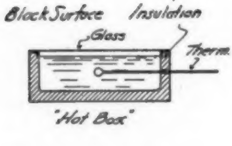

FIG. 1. HOT BOX OF DE SAUSSRE—1780

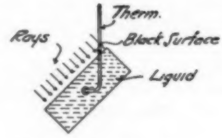

FIG. 2. POUILLET'S PYRHELIOMETER—1838

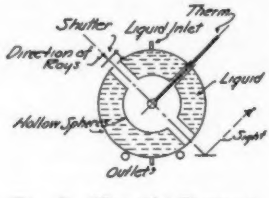

FIG. 3. VOILLE'S PYRHELIOMETER—1876

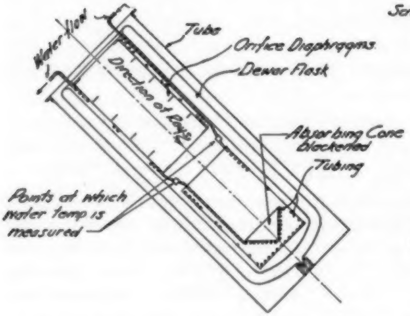

FIG. 4. SMITHSONIAN INSTITUTION STANDARD PYRHELIOMETER

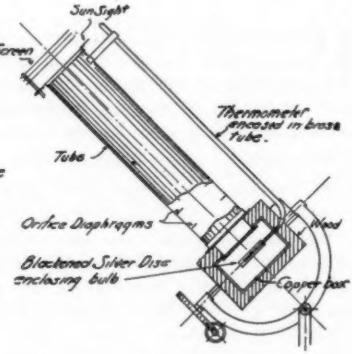

FIG. 5. SILVER DISC PYRHELIOMETER (G. C. ABBOT)

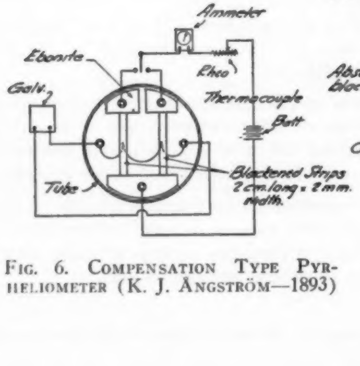

FIG. 6. COMPENSATION TYPE PYRHELIOMETER (K. J. ÅNGSTRÖM—1893)

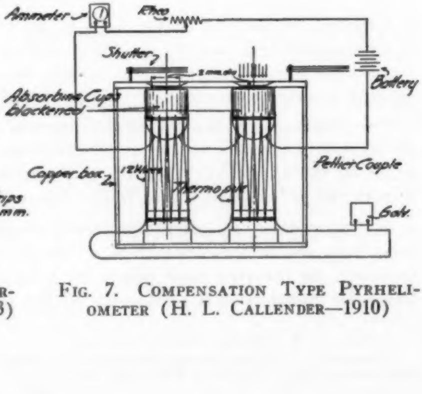

FIG. 7. COMPENSATION TYPE PYRHELIOMETER (H. L. CALLENDER—1910)

Classification of Instruments

No instrument can provide instantaneous readings of the rate at which total solar radiation is absorbed, on account of the element of time required for obtaining practically thermal equilibrium of the heat absorbing element and also due to the fact that the energy received is constantly varying in amount.

Instruments that have been devised for the measurement of the intensity of solar radiation may be classified, in general, according to the manner in which this measurement is effected. The five ways by which this has been accomplished are:

1. Recording the temperature rise of a liquid or solid substance of known specific heat in a given time.

2. Recording the temperature rise of a given weight of liquid, of known specific heat, in a given time used to remove the heat absorbed by a thin plate or inverted cone.

3. Recording the temperature difference on the two sides of a plate of known conductivity.

4. (*a*) A compensation method using the Joule effect by measuring the heat equivalent of electrical input to a plate, inverted cone or other type of absorber similar in character and dimensions to the heat absorber exposed to the sun's rays. The record is made when these two absorbers are in thermal equilibrium.

(*b*) A compensation method using the Peltier thermo-electric effect by means of which the heat is absorbed and removed as rapidly as it is received by the heat absorbing element.

5. Recording the temperature of a plate after thermal equilibrium has been established. This instrument must be either calibrated by the use of some other type or the combined surface coefficient for radiation and convection accurately known for the plate with the same range of the temperature difference between the plate and practically still air as may be encountered in practice.

A Frenchman, de Saussre, about 1780, constructed the hot box as indicated by Fig. 1, but it is not known that he made any attempt to estimate the actual amount of solar energy absorbed by the apparatus.

Class 1. The first instrument (Fig. 2) devised for actually estimating the intensity of solar radiation at the earth's surface was invented by the French physicist, Pouillet, 1838. Pouillet gave his instrument the Greek name *pyrheliometer* or "that which measures the fire of the sun." This name has been applied to all subsequent instruments devised for the purpose of measuring total solar radiation.

This type of instrument was somewhat improved by Voille in 1876 (Fig. 3), who placed the bulb of the thermometer at the center of two concentric spheres as shown. Water was circulated between the surfaces of the spheres to maintain a fairly constant temperature for the surface to which radiation from the thermometer bulb takes place.

Instruments in this class are sluggish in action and the correction for heat loss by radiation and convection is generally supposed to be somewhat uncertain. The heat absorbing body in the Voille instrument was the blackened thermometer bulb, the specific heat and weight of which was determined and later used in the calculation. This arrangement, doubtless, permitted obtaining

somewhat quicker and more consistent results than with the Pouillet instrument.

Instruments of this type are operated as follows:

(*a*) Observe the fall in temperature (θ_1) for time t when a screen is placed over the instrument.

(*b*) Remove screen and observe the rise in temperature (θ_2) for the same period of time t.

(*c*) Replace screen and observe the fall in temperature (θ_3) for time t.

The temperature rise (θ) that would have occurred for time t with no loss by radiation and convection is assumed to be:

$$\theta = \theta_2 - \frac{\theta_1 + \theta_3}{2}$$

The rate of solar energy received is therefore,

$$h = \frac{sw\theta}{At}$$

in which h = Btu, square feet, hour
s = specific heat, Btu, pounds
w = weight, pounds
A = area surface exposed to normal rays, square feet
t = time exposure, hours

A comparatively recent type of instrument (Fig. 5) based on the same general principle of operation was devised by G. C. Abbot, Secretary of the *Smithsonian Institution,* and is now largely used in this and other countries. The thermometer bulb is embedded in a hollow silver disc with the space between bulb and disc filled with mercury to insure positive contact between the heat absorbing body and thermometer. This instrument is provided with calibration constants which are determined for each individual instrument by comparing readings taken simultaneously, with the standard Smithsonian water flow pyrheliometer.

Class 2. The water flow pyrheliometer (Fig. 4) was devised by the staff of the *Smithsonian Institution* about 1910 and is considered the standard instrument by which other pyrheliometers in this country are compared and calibrated.

Briefly, the instrument consists of a heat absorbing thin metal blackened inverted cone surface located in a Dewar vacuum flask chamber at the lower end of a tube. The inverted cone is employed to eliminate the uncertainty of the cofficient of absorption which depends not only on the blackness of the surface, but also upon the wave length of the radiation received. The cone provides a black body absorbing surface that is said to give complete absorption with an error of less than 0.1 per cent. A plane surface painted dull black is generally assumed to absorb approximately 98 per cent of the radiation incident upon it.

The purpose of the tube, provided with the orifice plates, is to prevent convection air currents from removing heat from the cone surface. The sun's rays are kept normal to the plane of the cone base.

The underside of the cone and the periphery of the enclosing cylinder for a short distance above the cone is wound with a small diameter tube through which water is circulated. Provision is made for accurately measuring the temperature rise of the water passing through the coil and its rate of flow.

The intensity of solar radiation (h) may then be calculated. w = weight *of* water passed through apparatus in time t, pounds per hour; s = specific heat water corresponding to the average temperature; θ = temperature rise of the water, degrees Fahrenheit; t = length of exposure, hour; A = area of the aperture directly above cone through which the normal rays pass, square feet; h = intensity of solar radiation received, Btu, square feet per hour, or

$$h = \frac{ws\theta}{At}$$

Class 3. The only application of the heat transmission principle for the measurement of solar radiation known to the writer was made by F. C. Houghten in 1930 (Fig. 8). He employed a Nicholls heat meter developed by the A. S. H. V. E. Research Laboratory, which is described on pages 139-142, A. S. H. V. E. TRANSACTIONS 1930. The lower side of the meter was placed in direct contact with a water cooled surface and is reported to require approximately 10 min for thermal equilibrium to take place when the intensity of radiation is fairly constant.

It would appear that this class of instrument possesses the virtue of a fair degree of accuracy when the conductivity of the material composing the meter is definitely known for various mean plate temperatures and when the absorption coefficient for the exposed surface is well established.

Class 4. Compensation types.

(*a*) The first instrument of this class was devised by the Swedish physicist, K. J. Ångström in 1893. (See Fig. 6.) The heat absorbing elements, of similar dimensions (2 cm long $\times$ 2 mm wide), were constructed of thin strips of manganin metal. The electrical resistance of this strip of material is practically constant for various temperatures so that only the current, in amperes, need be measured in practice to determine the watts input.

One strip is exposed to the normal rays with the other shielded. An electric current is passed through the latter strip of sufficient strength to just equalize the temperature of the two strips and assuming equal heat loss by radiation and convection from the two strips, it is evident that the heat equivalent of the electrical input (Joule effect) to the first strip must be equal and is the measure of the heat equivalent of the solar energy received by the second strip.

A thermocouple insulated from but attached to the back of each strip, in circuit with a galvanometer, is employed to determine when the temperature is equalized between the strips.

(*b*) The instrument, shown in Fig. 7, and termed *radio balance,* was devised by the English physicist, H. L. Callender, in 1910.

The radiation received is totally absorbed by a small blackened sheet copper absorbing cup, where it is directly compensated by the absorption of heat due to the well known Peltier effect in a thermojunction formed between the cup and a constantan wire through which a current is passed.

Any change in temperature of the cup is indicated by a galvanometer connected to a thermopile composed of 12 wires (iron and constantan) which support the cup. The cold junctions are attached to the copper casing as indicated. The thermopile is electrically insulated from cup and casing by the use of thin silk paper and paraffin wax. The constancy of the temperature inside the box is indicated by a sensitive thermometer, not shown, placed between the

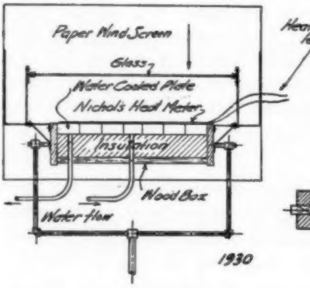

FIG. 8. CROSS-SECTION OF A. S. H. V. E. RESEARCH LABORATORY PYRHELIOMETER—1930

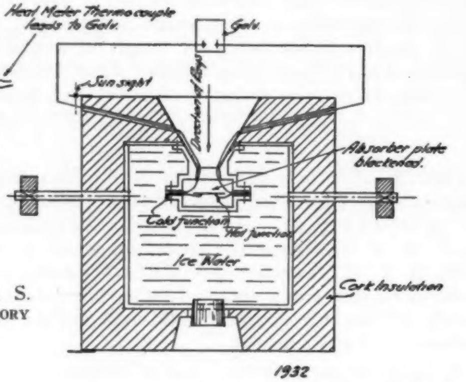

FIG. 9. CROSS-SECTION OF A. S. H. V. E. RESEARCH LABORATORY PYRHELIOMETER—1932

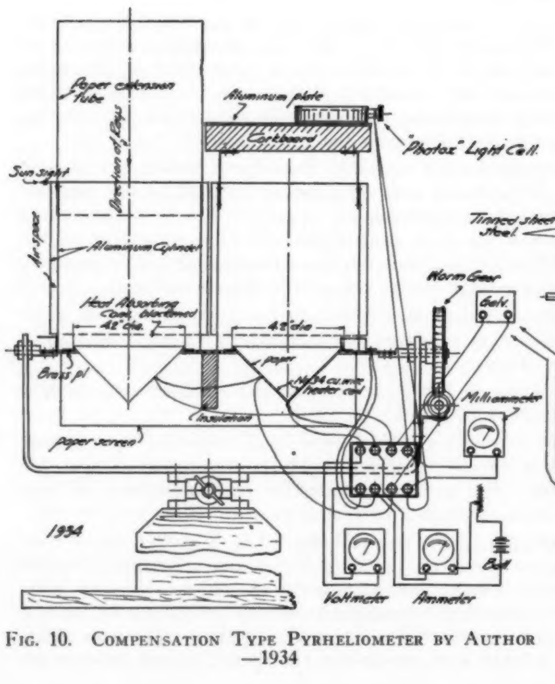

FIG. 10. COMPENSATION TYPE PYRHELIOMETER BY AUTHOR —1934

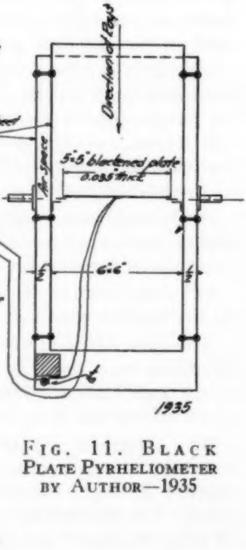

FIG. 11. BLACK PLATE PYRHELIOMETER BY AUTHOR—1935

cups. One cup is exposed to the normal rays and a measured current allowed to flow through the thermojunction of the couple attached to the cup.

The value of the Peltier coefficient P for a single copper-constantan junction is approximately 12 millivolts, which when multiplied by the current C in amperes gives the heat absorption PC in milliwatts.

With an aperture of 2 mm diameter, the current required to compensate for mean solar radiation (0.070 watt per square centimeter or 222 Btu, square feet) is about 200 milliamps with a single thermocouple. In practice the current is passed through the Peltier junction of both cups so that the exposed cup is cooled while the screened cup is heated. This doubles the effect and requires a current of 100 milliamps. The equation for the intensity (h) of solar radiation received and measured in milliwatts per square centimeter is:

$$h = 2PC/a \quad a = \text{area of aperture, square centimeters}$$

The value of P may be accurately determined by fitting a resistance coil in the cup and measuring the heating effect.

The author, adopting the principle used by Ångström, constructed the pyrheliometer in 1934 shown by Fig. 10. This instrument consists of two similar 45 deg inverted cones constructed with 0.035 in. thickness sheet brass. Each inverted cone is 4.2 in. diameter with a face area of 0.0962 sq ft. The cones are supported by thin cardboard attached to the brass plate as indicated and the plate movement is controlled by a worm and worm gear.

One cone is wound on the inside with No. 34 gage silk insulated wire, approximately ⅛ in. pitch winding and the winding is covered with thin paper shellacked in place. The cone surfaces are painted dull black inside and out with lampblack, shellac and alcohol.

The direct current for operation is supplied from two 6-volt automobile batteries, controlled by a rather finely wound rheostat. Current measurements are taken by means of a voltmeter and ammeter.

The heater element is screened from the sun's rays by means of a cork shield, as indicated, covered on top with sheet aluminum. A copper-constantan couple is soldered to each cone in circuit with a galvanometer; the galvanometer reading being zero, when the two cones are in thermal equilibrium.

A reading is obtained by first pointing the instrument toward the sun by means of the sight, then adjusting the current by means of the rheostat until the two cones are the same temperature. The current input, in watts, is recorded and the rate at which solar heat is received is then determined by the equation,

$$h = \frac{\text{watts} \times 3.415}{0.0962} = 35.4 \text{ watts, Btu, square feet, hour}$$

This instrument has been used at various times during the past two years with good success. It was recently used to determine the solar heat transmitted by a special type of heat absorbing glass, as referred to later in this paper. The size of this instrument was determined largely by the fact that it was desired to use standard types of voltmeters and ammeters provided with tenth scale divisions. The larger the heat absorbing element in this type of instrument, the less is the likelihood of errors in the final results obtained. The obvious objection to this instrument is the fact that the set-up is somewhat complicated by the amount of apparatus involved.

The calibration of the heater cone shown by Fig. 12 for 70 deg air is used as a general guide in checking field measurements. The relation between the volt-ampere readings is affected by the air temperature, as would be expected, due to the increase in resistance of the copper wire winding as the air temperature increases. This complication could be removed by employing manganin wire for the heater cone, in which case it would only be necessary to read one current measuring instrument (ammeter) as the resistance of the manganin wire is little affected by its temperature.

Class 5. An instrument of this type (Fig. 9) was devised by F. C. Houghten and associates in 1931 and used by the A. S. H. V. E. Research Laboratory staff for solar radiation determinations. This instrument, which is fully described in A. S. H. V. E. TRANSACTIONS for 1932, was calibrated by means of the Abbot pyrheliometer.

The small heat absorbing plate is provided with two thermocouples, the cold

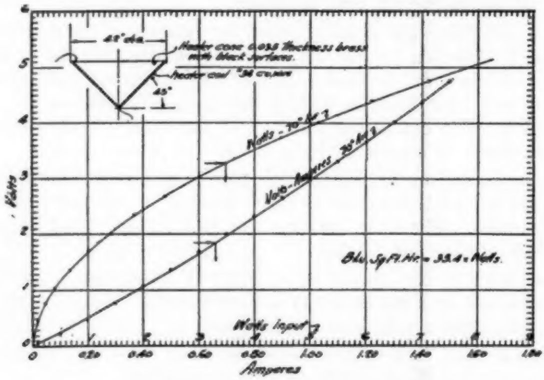

FIG. 12. CALIBRATION CURVES FOR HEATER CONE

junctions of which are located outside of the space surrounding the plate.

A reading of the galvanometer indicates the temperature differences between the heat absorbing plates and its surrounding enclosed space maintained constant with ice water.

This instrument possesses the advantage of a quick reading and consequent determination of the solar energy being received.

Description of Black Plate Instrument

The black plate pyrheliometer shown by Fig. 11 was constructed by the author in 1935 in an attempt to devise a simple type of apparatus which may be readily constructed and operated. It has been used to a limited extent with apparently satisfactory results.

The heat absorbing plate consists of a 5 in. by 5 in. brass plate 0.035 in. in thickness blackened both sides. The plate is supported in a housing as indicated and the inner surface of the inside enclosure is painted black. The two concentric enclosures are constructed of bright tinned sheets.

A copper-constantan thermocouple is soldered to the plate, the cold junction being soldered to a thin sheet brass strip enclosing the thermometer bulb, screened from the plate and sun's rays as indicated.

A portable type galvanometer, provided with a scale, is used to indicate the temperature difference between the plate and the air.

It is obvious that the heat absorbing element must be protected against air currents and the combined coefficient of radiation and convection (k) be carefully determined in comparatively still air. Four different electrically heated plates were constructed each 5 in. by 5 in. but of various materials, thicknesses

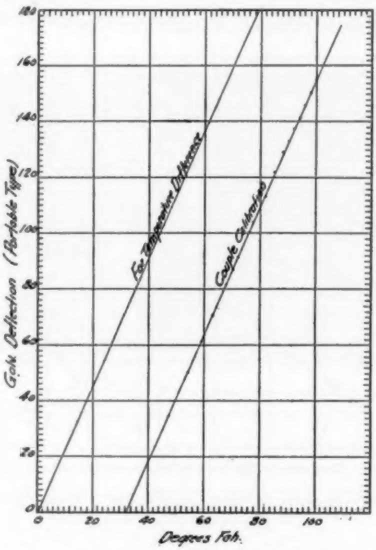

FIG. 13. CALIBRATION OF HEATER AND SUN PLATE THERMOCOUPLE

and copper wire resistance windings. The outside surfaces of the plates were painted dull black with lampblack mixed with shellac and alcohol.

The watts input to the heater were determined in the customary manner and the value of k determined with the heater plate in the position to be occupied by the heat absorbing plate in the open end enclosure. The results of several hundred observations obtained with the various plates checked quite closely.

Heater Plate Results

The results were plotted with divisions for the ordinates of 1 Btu, square foot, hour and ½ F temperature difference for the abscissa.

Fig. 14 gives the results of the heater plate tests. The two curves for k and h indicate the variation obtained for various temperature differences be-

tween the plate and air. The heat emitted per square foot per hour (test results) for various temperature differences is given by the equation,

$$h = (t_p - t_a)^{1.15}$$

and the heat emitted per square foot per hour per degree difference by the equation,

$$k = (t_p - t_a)^{0.15}$$

in which t_a = temperature air and t_p = temperature plate degrees Fahrenheit.

The writer is of the opinion that the calibrated blackened thin plate type pyrheliometer, as described, is sufficiently accurate for practical use.

The only instrument required is a portable type galvanometer provided with

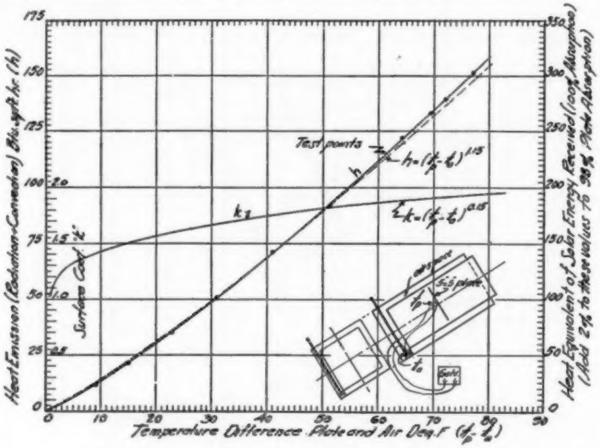

FIG. 14. CALIBRATION CURVES FOR 5 IN. x 5 IN. BLACK SURFACE PLATE IN ENCLOSURE

scale and a suitable shunt. The accuracy of the combination depends largely upon the accuracy of the original calibration, prevention of strong currents of air from entering the open ended enclosure and an allowance (98 per cent) for reflected rays. The latter allowance may be eliminated, if a calibrated cone were employed in place of the flat type plate absorber.

The calibration of the plate should be made in the same position as that in which it is to be used. The plate calibration, by the author, was made with the enclosure placed at an angle of 40 deg with the horizontal. There appears to be little or no difference in calibration between a 40 deg and a 15 deg angle.

Incidentally, this appears to be the general form of equation for the heat transfer from pipes to still air. The author recently tested a blackened ¼ in. brass pipe 3 ft. 0 in. long, suspended in still air, electrically heated, with the result.

$$h = (t_p - t_a)^{1.20} \text{ and } k = (t_p - t_a)^{0.20}$$

The author is not in a position to vouch for the preceding equations beyond

the temperature differences and a comparatively low range of temperatures used in these tests.

A fairly close check (averaging 1 per cent) was obtained in the laboratory between instruments as shown by Figs. 10 and 11 using the heat and light emitted by a 250 watt lamp. No direct comparative field tests have been made between the two instruments.

Measurement of Solar Radiation Through Glass

These instruments were used during the past summer in Buffalo, N. Y., to determine the transmission of solar radiation through a pane of heat absorbing glass, manufactured in England and having a thickness of ⅜ in.

The averages of a considerable number of tests obtained with this glass

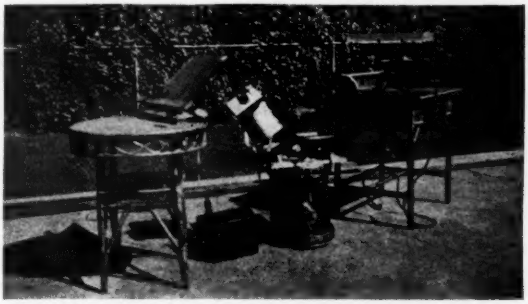

FIG. 15. ARRANGEMENT OF COMPENSATION TYPE PYRHELIOMETER USED FOR HEAT ABSORBING GLASS TESTS

were as follows. The percentages refer to 100 per cent solar energy received with rays normal to the glass and the heat absorbing elements.

Heat absorbed and reflected	77 per cent
Heat transmitted	23 per cent

When glass is used in building construction, approximately one-half of the heat absorbed by the glass may be assumed to also enter the interior of the building by radiation and convection from the hot glass surface. The estimated heat gain for the heat absorbing glass on the basis of this assumption would be: $38.5 + 23$ or 61.5 per cent. Actually the percentage would probably be somewhat less due to the higher temperatures of the outside surface of the glass.

The same kind and thickness of glass was tested, in Pittsburgh, Pa., by F. C. Houghten at about noon October 31, 1935. Two readings, taken approximately 20 min apart, resulted in the same percentages previously reported. The maximum heat received was 245 Btu per square foot per hour. The pyrheliometer used by Mr. Houghten in this test is shown by Fig. 9.

The following is a typical set of readings taken by the author with the black plate pyrheliometer (Fig. 11) and recorded in Buffalo, N. Y., on November

24, 1935. The readings shown were taken at 12:15 and 12:20 p. m., Eastern Standard Time.

ANGLE OF RAYS WITH HORIZONTAL, APPROXIMATELY 31 DEG, AIR TEMPERATURE 37 F

	No Glass	Heat Absorbing Glass
Galvanometer deflection	139	39
Temperature difference plate and air degrees Fahrenheit ($t_p - t_a$)	61	17.2
Btu, square feet, hour received (assumed 98 per cent plate absorption)	234.6	55
Per cent heat transmitted	...	23.4 per cent
Per cent heat absorbed and reflected	...	76.6 per cent
Photox cell—milliamps	8.8	5.2
Per cent light transmitted	...	60.0 per cent

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DISCUSSION

R. P. Cook (Written): My knowledge of the subject is totally inadequate to the preparation of a discussion which would be of any value to the society; however, I do wish to call the attention of the society to the tremendous importance of that factor which we call *Sun Effect* and to our extreme lack of accurate information on the subject. The measurement of solar radiation is relatively simple as compared to the problem of determining the final effect on conditions within a structure. Before this final effect can be computed accurately we must know, not only the intensity of the solar heat and the angle between the heat receiving surface and the sun's rays, but must also be able to compute the amount of heat reflected back into space, the amount absorbed by the material, the amount dissipated to the outside by convection and radiation, the amount actually carried through to the enclosure and the rate at which it is carried through. Not infrequently, in an air conditioning problem we find the sun effect equal to all the other heat gains combined and yet the determination of this heat gain must be based on less adequate information than any of the other factors.

This is not a criticism of the progress which has been made by our society in this connection, but rather an appeal to extend our efforts to arrive at a more logical and satisfactory method of computing sun effect. A number of excellent papers on this subject have been presented within the last few years, and considerable progress has been made by the research connections of our society. However, there is still an immense amount of work to be done in the matter of obtaining, collecting and formulating the information on this subject which is necessary to accurate design.

K. A. Powell (Written): It seems to me that this paper does not indicate the ratio of the total heat transmitted to the visually effective light transmitted, though it would seem to refer to that. Neither does it indicate the wave length energy distribution of the particular sun radiation under which the selective transmission tests were made.

The best approximation that I have been able to make of the relative heat capacity of translucent materials during some experiments in 1933-34 required the calibration of transmission by wave bands. The ratio of this band transmission, multiplied by the product of a chosen wave length energy distribution and the wave length visual sensitivity, to the product would give the light transmission efficiency. The band transmission multiplied by the energy distribution would give the heat transmission efficiency. The heat transmission efficiency divided by the light transmission efficiency should give the true percentage of translucent heat resistance efficiency.

The energy distribution in the sunlight seemed to vary considerably with meteorological, dust and smoke conditions, according to my observations. Also, the total energy generally varied too fast for instantaneous measurement with anything more sluggish than a Coblentz style thermocouple.

I would like to ask Mr. Harding by what means an overall wave length measurement in sunlight can be made to give even approximate heat to light selectivity calibration. Also, what constituent other than ferrous oxide was used in the English glass to confer infra-red opacity.

L. A. Harding (Written): The question of total solar radiation transmission test of the heat absorbing glass referred to by K. A. Powell was given to illustrate a common use of the pyrheliometer as indicated by previous Society papers. The light transmission test was incidental.

Perhaps the "per cent heat transmitted" and "per cent absorbed and reflected" should read, per cent total solar energy transmitted, etc. As now referred to in the paper, it may cause some confusion in the mind of a reader.

Air conditioning and refrigerating engineers are not concerned with a band division of solar energy obtained by making a traverse of the solar spectrum. The heat to be removed by the refrigerating apparatus is obviously the total solar energy passing through the glass and is customarily determined by means of the pyrheliometer.

The photronic cell is now generally employed to determine light intensity and the percentage of reduction in the intensity due to dirty and dust covered glass, etc. The copper oxide photronic cell, known as the *photox* cell, possesses the very desirable characteristic of a color response curve that compares very closely with the sensitivity of the average human eye under all wave lengths of visible light. The cell is not responsive to wave lengths on either side of the visual band. There are, however, other cells on the market which are somewhat responsive to wave lengths outside of this band. The current-light intensity curve varies with the external resistance in circuit with the cell and are not straight lines at least in the lower intensities (below 500 ft candles).

It is necessary to use a screen with the cell and micro ammeter combination which make up the light meter when employing same in strong light.

The percentage of light passed by the heat absorbing glass, as stated, was determined from the readings of a milliammeter in circuit with an unscreened cell. About the same percentage was observed by means of the light meter employing a white paper screen for the cell.

No great claims are made for the accuracy of the value stated (60 per cent) and I would be inclined to use a figure of 55 per cent as a fair average for this particular glass.

It is, quite obviously, impossible to state a ratio of total energy to energy in the visual band of the spectrum by means of a pyrheliometer or by the combined application of this instrument and a photronic cell. No such ratio is inferred by the values stated in the paper.

The purchaser and user of a heat absorbing glass is interested in the reduction in solar energy entering the building through windows and skylights when this glass is employed, as compared with the various commonly used types (plain, ribbed, etc.).

The intensity of daylight illumination will obviously be reduced, but as shades are customarily employed or the usual types of glass windows whitewashed any very exact value for the reduction in the daylight illumination by the use of this glass is relatively unimportant.

The paper states that the figures given refer to rays normal to glass with clear sky. The percentages naturally vary somewhat with rays other than normal, variations in the selective absorption of the earth's atmospheric envelope and sun angle.

With normal rays, clear sky and sun angle of approximately 30 deg, the following average percentages of the total solar radiation, for the three mentioned bands, were calculated from Smithsonian data—Mt. Wilson Observatory Station.

BAND	WAVE LENGTH μ	PERCENTAGES
Ultra violet	0.00 — 0.45	12
Visual	0.45 — 0.70	40
Infra red	0.70 — ∞	48
		100

Applying these data to the tests mentioned in the paper and the fact that glass absorbs practically all of the ultra violet rays, the following conclusions result.

BAND		BTU PER SQUARE FOOT PER HOUR
Ultra violet	=	0.0
Visual 235 x 0.40 x 0.55	=	52.0
Infra red (by difference)	=	3.0
Total passed as measured by pyrheliometer	=	55.0

These values would infer the ratio mentioned by Mr. Powell, and would appear to indicate that the percentage of infra-red rays passed by this particular glass is relatively small.

As one would infer from the test, this glass, when exposed to the sun's rays with a high dry-bulb air temperature, becomes very warm, in fact quite hot. The glass, however, did not crack due to unequal expansion, under the maximum temperature conditions existing last summer in this vicinity. It is anticipated that there will be a demand for a heat absorbing glass, when it becomes generally known that such glass is obtainable.

In conclusion, I am under the impression that whenever a selective band traverse of the solar spectrum is made by the Smithsonian staff, at their various stations, the pyrheliometer is the instrument depended upon to furnish the summation of the total energy. The traverse apparently requires about seven minutes with the highly developed photographic recording devices employed by this institution, whereas the pyrheliometer (Abbott silver disc type) observation requires about one minute.

K. A. Powell (Written): If it is contended by Mr. Harding that selective transmission does not concern his problem, then the use of glass may be questioned if visible wave length transmission is not important. Certainly the cheapest way to reduce total sun radiation transmission is to reduce the translucent area with an opaque reflector.

The use of the copper oxide *photox* cell to measure the proportion of visible radiation transmitted, to compare with the proportion of total radiation transmitted, had occurred to me; but the problem is then to maintain a constant wave length distribution in the source during the comparative tests.

A Nernst glower source, operated on a storage battery, and a Hilger rock-salt-prism monochromatic illuminator permit close duplication of tests without relation to meteorological conditions. The Coblentz thermopile and galvanometer arrangement is rapid enough to complete any given wave length observation in less than a minute.

I have used Corning Glass Company wave length screens to check the effectiveness of the Hilger spectroscope. The use of such screens would not permit the spectroscope to be eliminated altogether, unless one could obtain a constant artificial source of sun wave-length energy distribution. Since completing my last work, I understand that a sun energy distribution source is available.

I was especially interested, while reviewing my work, to note the possibilities of the spectroscopic method with a sylvine prism, for investigating selective energy transmission, reflection and absorption phenomena at the predominant black body radiation wave lengths of source temperatures down to freezing. The importance of this was brought out by some rough experiments supervised by Prof. F. B. Rowley, in which relative insulation values reversed under different radiator temperatures. Also a test which I made showed a visibly opaque coating of Prussian blue to be more transparent in the infra-red range than good window glass is in the visible range.

It has been suggested that there is a prohibitive expense of monochromatic equipment for use in heating and ventilating work. A survey which I made three years ago indicated that a suitable test equipment could be assembled for about $1200.00, which is not unreasonable for the work it will accomplish. Most college physics departments have the necessary equipment.

R. A. Miller: In presenting his paper on Pyrheliometry, Mr. Harding has made a very definite contribution to the literature of the Society and has certainly performed a service of real value in condensing into a single paper a concise summary of the art. It is, therefore, hardly proper to offer any serious discussion of the general subject which Mr. Harding has so thoroughly covered.

However, it seems to me important to at least suggest some further consideration of the problem of heat transfer through heat-absorbing glass, since the results which Mr. Harding has obtained with his English glass would appear to be at variance with results obtained when American glass of less thickness has been tested. These differences are more apparent than real and would seem to be virtually wholly due to the differences in thickness of the samples measured, rather than to any difference in the actual effectiveness of the glasses themselves.

In his discussion of the total heat gain to the interior of a building through the English glass, ⅜ in. thick, Mr. Harding has included the value of the heat reflected from the first surface which, of course, would never have any possibility of entering the building. Deducting this value of 4.5 per cent from the total solar heat available, and also deducting the 23 per cent shown as actually transmitted, leaves available 72 per cent for re-radiation. On the assumption of 50 per cent re-radiation to the interior and 50 per cent re-radiation to the exterior, the total heat gain would be 59 per cent of the incident solar energy. If the values for the English glass are converted to a ¼ in. thickness, an actual transmission value of 36.5 per cent instead of 23 per cent is obtained. This gives a re-radiated energy value to the interior of the building of 29.5 per cent. The sum of the re-radiated energy plus the actually transmitted energy is 66 per cent of the total incident energy. The visible light transmission shown for the ⅜ in. thick glass is 60 per cent and converting this to ¼ in. thickness, 70 per cent total light transmission is obtained.

Considering the American glass products of ¼ in. thickness, which are readily available from several different sources, and using the same tabulation as before, an actual transmission value of 43.5 per cent plus a re-radiation value of 26 per cent is obtained, giving a total energy gain to the interior of the building of 69.5 per cent, which for purposes of comparison may be considered 70 per cent. The total visible light transmission of this glass is found to be 79 per cent.

These comparisons are not strictly accurate, especially when the measurements are made through heat-absorbing glasses. Solar energy arriving at the earth's surfaces is constantly changing from day to day and from moment to moment, due to variations in the water vapor content of the air and the corresponding absorption of rays of different lengths. The effects of water vapor upon visible light radiation are much less marked than are those same effects in the infra-red portion of the solar spectrum and consequently the visible light transmissions are more nearly comparable than are the heat-absorptions or heat transmission values. However, on the assumption that the conditions under which the measurements were made were exactly identical, the differences shown in heat absorption are of a very small order of magnitude and the differences in light transmission are essentially insignificant.

It is also interesting to note that there are available other American glasses, which, in ¼ in. thickness, will transmit approximately 36 per cent of the total solar energy and only approximately 53 per cent of the total visible light.

The essential point to be brought out by this discussion is the fact that satisfactory American glasses of almost any desired characteristics are available and that we do not need to turn to our English neighbors for useful and thoroughly satisfactory products. I would also like to emphasize that the determination of the actual values of heat-absorbing glasses by Pyrheliometry may be decidedly unreliable because of conditions entirely beyond the control of the investigator. Consequently, the evaluation of such glasses should be entirely on a laboratory basis where conditions can be constantly controlled and a reading made one day can be definitely checked on any subsequent day and the values for one glass are entirely comparable with the values for another.

No. 1044

A FIELD STUDY OF THE HEAT REQUIREMENTS OF A COLLEGE BUILDING

By F. E. GIESECKE * AND W. H. BADGETT ** (*MEMBERS*)
COLLEGE STATION, TEXAS

This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in cooperation with the Texas Agricultural and Mechanical College.

IN cooperation with the Research Laboratory of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, the Texas Engineering Experiment Station of the Agricultural and Mechanical College of Texas conducted a field study of the heat requirements of the Petroleum Building at the college, during the heating season of 1935-1936.

The building used for these tests, shown in Fig. 1, houses the Texas Engineering Experiment Station, the Department of Petroleum Engineering, and the Department of Geology. The building has a reinforced concrete frame, and walls of brick, tile, and cast stone. Its cubage is approximately 522,000 cu ft and, with respect to heat storage capacity, its mass is approximately equivalent to 1,700,000 lb of water, not including the tower or the foundations. The tower houses the expansion tank for the central hot water heating plant which supplies heat to 38 major college buildings. The expansion tank has a capacity of 8000 gal.

The building under test is heated by a reversed-return hot water system having 9500 sq ft of equivalent direct cast iron radiation. Heat is supplied to the building continuously, day and night, except during those periods when the outdoor temperature is above 65 deg. Water for the system is supplied from the central mains in the heating tunnel located in front of the building. By employing the valve and pump equipment shown in Fig. 2, the hot water from the power plant is mixed with cooler water from the return main in the building to secure the desired flow temperature in the system, and then pumped through the various radiator circuits in the building. This mixing and recirculating arrangement is employed to permit the use of higher temperature water

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in the tunnel flow mains and a correspondingly smaller volume of water, and to provide for the variations in the individual requirements of the many buildings heated from the central plant. Once a satisfactory setting of the mixing valves is obtained for a building, it is not disturbed for the remainder of the heating season. The water temperature in the flow mains of the tunnel is controlled at the power plant and is varied according to the outdoor temperature as shown in the accompanying schedule:

Outdoor Temperature	Flow Main Water Temperature	Return Main Water Temperature
25	240	180
30	221	168
35	202	156
40	183	143
45	165	131
50	146	119
55	127	107
60	109	94
65	Heat Discontinued	...

Electrical recording and integrating flow meters were installed in the supply and return mains of the building, as shown in Fig. 2, to determine the quantity of water which passed through the heating system of the building. These meters were calibrated in the laboratory of the Texas Engineering Experiment Station before they were installed and their percentages of error over the entire operating range were found to be less than plus or minus 2 per cent. After installation, the operation and accuracy were checked by a factory representative. The temperatures of the water in the supply and return mains were registered by a two-pen recording thermometer which was very frequently checked against calibrated indicating thermometers located in the mains adjacent to the recording bulbs (Fig. 2). From the data provided by the recording flow meters and the recording thermometers the quantity of heat dissipated by the heating system of the building during any period of time could be readily calculated.

Outdoor temperatures were secured from a recording thermometer located in a standard United States Weather Bureau thermometer shelter 100 yards north of the building. This instrument was checked at frequent intervals with a Bureau of Standards calibrated thermometer.

Indoor temperatures were taken at representative points throughout the building from 30 wall-type indicating thermometers and 10 recording thermometers placed 5 ft above the floor, on the inside walls and away from windows.

Wind velocities were recorded by an electrical wind velocity recorder of the U. S. Weather Bureau pattern with a standard three-cup anemometer located 9 ft above the roof of the building at an unobstructed point. The direction of the prevailing wind was determined by observing an indicating weather vane also located on the roof. With the exception of the south side, the building was not sheltered in any manner from the effect of the wind.

In computing the heat requirements of the building, the procedure and recommendations in the 1935 edition of THE A.S.H.V.E. GUIDE were followed with extreme care. From studies of temperature and wind velocity records of the U. S. Weather Bureau observation station at the college, an outdoor design temperature of 25 F and a wind velocity of 15 mph from north to northwest were selected as meeting THE GUIDE requirements.

An indoor design temperature of 70 F was used for the entire building except for laboratories of a shop nature and for halls for which a temperature

FIG. 1. TEXAS ENGINEERING EXPERIMENT STATION, PETROLEUM ENGINEERING AND GEOLOGY BUILDING, AGRICULTURAL AND MECHANICAL COLLEGE OF TEXAS

of 65 F was used. The temperature beneath concrete floors on a fill was assumed to be 5 deg below the room temperature.

In the earlier calculation, the air temperature beneath the floor slab suspended above the ground was assumed to be 50 F. After several weeks of observations, a recording thermometer was placed in several locations beneath the slab for a period of two weeks. During this time the outdoor temperature range was from 15 F to 65 F, whereas, the air temperature beneath the floor slab varied only from 69 F to 71 F. The heat required to maintain a temperature of 70 F beneath the floor slab was supplied by the insulated flow and return mains, and by the riser branches beneath the slab, and should, therefore, be included in the estimate. Consequently, the heat losses for the foundation walls enclosing the space beneath the suspended floor slab were calculated and substituted for the previously determined losses through the floor slab.

No empirical exposure factors were used to increase the calculated heat losses on the sides of the building exposed to the prevailing winds. Air

leakage was determined by the crack-infiltration method from the values given in Table 2, Chapter 6, of THE 1935 GUIDE.

For the conditions outlined above, the heat requirements of the building were estimated to be 1,399,000 Btu per hour, or 31,750 Btu per hour, per degree difference, indoor to outdoor temperature.

The results of calculations for observations made during seven complete 24-hour periods, from midnight to midnight, in January and February, 1936, indicate that the estimated heat requirement averaged about 23 per cent higher than the actual heat requirements of the building. The estimated requirement

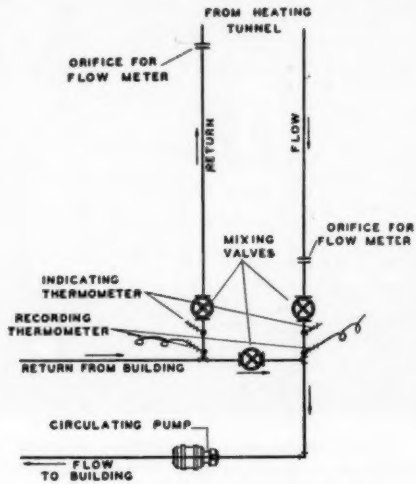

FIG. 2. METHOD OF MIXING AND RECIRCULATING WATER

was higher than the actual requirement for the seven 24-hour periods by the following percentages: 24.6, 28.7, 29.2, 21.4, 14.9, 28.8, and 17.2 per cent.

The general character of these 24-hour observations is shown in Table 1, which contains the observations for the 24-hour period of January 27, 1936.

The estimated and the actual heat requirements were determined in the following manner. From the average hourly indoor temperature, secured from the 40 thermometers distributed throughout the building, the average hourly outdoor temperature was subtracted. This difference, multiplied by 31,750 Btu, the estimated heat requirement per degree-hour, gave the estimated hourly heat requirement. The hourly heat requirements were then totaled for each 24-hour period.

From the flow meter records the number of pounds of water flowing through the heating system each hour was determined and this, multiplied by the

average hourly difference in the temperature of the water in the flow main and of that in the return main, for the same hour, gave the quantity of heat delivered to the building during that hour. The hourly quantities of heat delivered were then totaled for each 24-hour period.

The observations reported in this paper were all made on totally cloudy

TABLE 1. OBSERVATIONS FROM MIDNIGHT TO MIDNIGHT, JANUARY 27, 1936

Average Building Temperature 73 F

Hour of Day	Out-door Temp. F	Degree Hours	Water Temperatures F			Lb Water Flowing per Hour	Heat Supplied Btu	Wind Vel. Mph and Direction	General Weather Condition
			Flow	Return	Difference				
A.M.									
M–1	37.1	35.9	191.0	143.0	48.0	19,500	936,000	15N	Cloudy
1–2	37.0	36.0	192.5	144.0	48.5	19,920	966,120	15N	Cloudy
2–3	36.8	36.2	194.0	146.0	48.0	20,520	984,960	15N	Cloudy
3–4	35.8	37.2	196.7	148.0	48.7	21,120	1,028,544	15N	Cloudy
4–5	34.0	39.0	202.8	150.0	52.8	21,000	1,108,800	11N	Cloudy
5–6	32.7	40.3	206.4	153.0	53.4	20,820	1,111,788	9N	Cloudy
6–7	32.6	40.4	211.0	155.0	56.0	20,880	1,169,280	11N	Cloudy
7–8	32.0	41.0	208.6	155.0	53.6	20,520	1,099,870	13N	Cloudy
8–9	32.0	41.0	205.0	153.0	52.0	18,960	985,920	13N	Cloudy
9–10	31.8	41.2	202.0	149.0	53.0	18,720	992,160	14N	Cloudy
10–11	31.9	41.1	200.3	147.0	53.3	18,780	1,000,970	15N	Cloudy
11–N	32.4	40.6	200.0	146.0	54.0	18,540	1,001,160	11N	Cloudy
P.M.									
N–1	32.8	40.2	201.0	146.5	55.5	18,780	1,042,900	12N	Cloudy
1–2	32.8	40.2	202.0	147.8	54.2	18,600	1,008,120	12N	Cloudy
2–3	32.4	40.6	204.2	148.4	55.8	18,600	1,037,880	11N	Cloudy
3–4	32.0	41.0	207.0	150.2	56.8	18,300	1,039,400	9N	Cloudy
4–5	33.0	40.0	205.0	151.0	54.0	18,600	1,004,400	7N	Cloudy
5–6	33.5	39.5	204.2	149.3	54.9	18,420	1,011,260	9N	Cloudy
6–7	33.3	39.7	207.8	150.5	57.3	18,300	1,048,590	5N	Cloudy
7–8	33.0	40.0	206.5	151.3	55.2	18,300	1,010,160	6N	Cloudy
8–9	33.1	39.9	209.0	151.2	57.7	18,600	1,073,220	6N	Cloudy
9–10	33.1	39.9	210.6	153.0	57.6	18,600	1,071,360	4N	Cloudy
10–11	33.2	39.8	210.4	154.0	56.4	18,600	1,049,040	3N	Cloudy
11–M	33.0	40.0	209.0	154.0	55.0	18,840	1,036,200	3N	Cloudy

Total 950.7 × 31,750 = 30,184,700 Btu
Total Btu supplied = 24,818,090
Average = 10.2 Mph

days. In this manner the effect of solar radiation was eliminated from the calculations. Weather conditions during these tests were as near design conditions as could be secured: that is, an outdoor temperature of 25 F and a wind velocity from the north of approximately 15 mph. There was no precipitation in any of the cases reported, except a very light snow on the morning of February 4.

On days when the sun was shining, the effect was immediately discernible in the rooms exposed to the sun. The temperature in those rooms rose rapidly to as much as 10 deg above that when the sun was not shining, while the temperature in the rooms on the shaded portion of the building remained normal. This clearly indicates that automatically controlled radiator circuits, zoned for the various exposures of a building, are very desirable.

After these studies had been made and the per cent of variation between the estimated and the actual heat requirements of the building found to be approximately 23 per cent, an additional check was made of the calculations

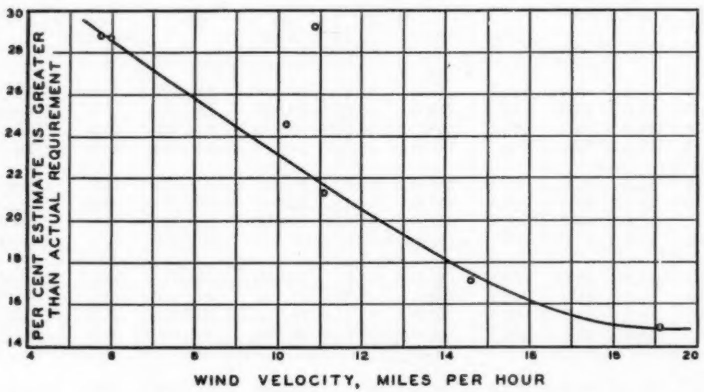

FIG. 3. CURVE SHOWING PER CENT ESTIMATED HEAT REQUIREMENTS ARE GREATER THAN ACTUAL PLOTTED IN RELATION TO WIND VELOCITY

used to estimate the heat requirements of the building. With a thickness gage, the cracks between the steel sash and the window frame were measured on a great number of windows of the building and found to be considerably smaller than those assumed. Consequently, the infiltration of air around the windows was corrected to more nearly conform with the actual crack found to exist for the windows in the building. This correction reduced the total estimated heat requirement under design conditions to 1,271,700 Btu per hour, or 28,900 Btu per degree difference, indoor to outdoor temperature. Thus, the estimated heat requirements were reduced to 13.5, 18.3, 17.7, 10.5, 4.6, 17.3, and 6.5 per cent higher than the actual heat requirement, or an average of 12.6 per cent more estimated than required.

In Fig. 3 the average wind velocity per hour for each of the seven 24-hour periods reported was plotted against the percentage the estimated heat requirements were higher than the actual heat consumed. Although more data are desirable, a definite trend is indicated in the relation between wind velocity and percentage of variation between the estimated and the actual heat requirements of the building.

From this study it appears that the greatest source of error in making an estimate of the heat requirements of a building occurs in allowances made for air infiltration. A calculation shows that if the air change method, based upon Table 3, Chapter 6, of THE 1935 GUIDE, were followed, even allowing only one air change per hour, the heat requirement estimate would be considerably larger than indicated above.

DISCUSSION

W. W. TIMMIS: Over the past three years I have made a rather intensive study of the heating requirements of buildings, particularly with reference to the proper distribution to the various rooms of buildings. In this work I have developed a very accurately calibrated radiator inlet valve. This valve can readily be set to supply a given quantity of steam at a given pressure differential. The application of these valves to a large number of buildings, particularly residences, gave us an unique opportunity of observing the correctness of our methods of estimating heating requirements particularly with reference to correct distribution. The heating requirements for these buildings were carefully figured in accordance with various accepted methods. The calibrated orifice valves were then set to supply steam at a rate corresponding to the condensing capacity of each of the radiators. Thus the valve on a 40 sq ft radiator would be set to supply 10 lb of steam per hour, on a 60 ft radiator 15 lb of steam per hour, etc. The heating systems in these buildings were controlled so as to supply steam in accordance with weather demands to maintain uniform indoor temperatures. It was found in almost every case that the distribution was not correct, although in most cases the total was correct, and in this regard our findings confirm those of the authors of this paper. The fact that the distribution was incorrect was indicated by the fact that the rooms were not all at the same temperature within even a reasonable range. In many cases means were provided to change the setting of the orifice valves so that the temperature in all the rooms was the same. This was possible only when no rooms in the building were under-radiated, and it was accomplished by cut and try methods. After the system had been balanced it was possible to go back with a special gage made for the purpose, and read the setting which had been determined by the cut and try method of balancing the system. By making such observations in quite a large number of installations it was possible for us empirically to arrive at a method of estimating heating requirements which appears to give more satisfactory results.

What has been said in the previous statements applies particularly to residences. Our observations in larger buildings such as apartment houses, office buildings, etc., are not nearly so complete but so far they seem to indicate that distribution by present methods of estimating heating requirements is reasonably satisfactory. In residences we found in general that the second floor of a two-story house is considerably under-radiated in relation to the first floor. We also found that in general living rooms are over-radiated in proportion to the rest of the house.

W. C. RANDALL: I am particularly interested in the statement in the last paragraph that "It appears that the greatest source of error in making an estimate of the heat requirements of a building occurs in allowances made for air infiltration." This would seem to indicate that the introduction in THE GUIDE 1936 of infiltration values for 1/64 in. crack to supplement those for 1/32 in. crack was justified.

I would like to ask the author if data are available as to the average crack found between window frame and movable ventilator.

It would appear from the values determined from the computed heat savings that the average crack might be less than even the 1/64 in. recommended for good manufacture and installation. In any event, the results indicate that the revisions in THE GUIDE 1936 were justified. The average crack between the frame and ventilator, as found on these steel casements, would be a contribution to a conservative basis of figuring heat losses, due to infiltration.

No. 1045

HEATING REQUIREMENTS OF AN OFFICE BUILDING AS AFFECTED BY WEATHER CONDITIONS

By F. C. Houghten * (*Member*) and Carl Gutberlet ** (*Non-Member*)
Pittsburgh, Pa.

This paper is the result of research sponsored by the American Society of Heating and Ventilating Engineers in its Research Laboratory at the Pittsburgh Experiment Station of the U. S. Bureau of Mines

HEAT requirements of buildings have been the subject of intensive study by the Research Laboratory and cooperating institutions for several years. This investigation has included studies of the overall heat requirements of various groups of buildings calculated from the steam supplied by a central station, the heat requirements of individual buildings based on their steam requirements, and more intensive studies of the heat requirements of individual rooms in a building. This work has been carried on through 1935 under D. S. Boyden, then chairman, and more recently under O. W. Armspach, the present chairman of the Technical Advisory Committee on Heat Requirements of Buildings.

Included in this general investigation was an intensive study of the heat loss from eight rooms on three different floors and three different exposures of the Grant Building, Pittsburgh. The observations were made during the latter half of the 1933-34 heating season. A portion of the results were immediately analyzed and a report,[1] dealing primarily with the relation between wind velocities as reported by the Weather Bureau and as observed near the building, was presented to the Society at the 1934 Semi-Annual Meeting. This present report is based on further analysis of the test observations and deals with the relation between the heat requirement of the several rooms and the weather conditions, including outside temperature, wind velocity and direction, and solar radiation. The effect of infiltration due to wind velocity, temperature difference and elevation in the building will be only briefly discussed in this paper. A more comprehensive analysis of this phase of the subject will be presented in a future report.

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* Director, A. S. H. V. E. Research Laboratory.
** Research Assistant, A. S. H. V. E. Research Laboratory.
[1] Wind Velocities Near a Building and Their Effect on Heat Loss, F. C. Houghten, J. L. Blackshaw and Carl Gutberlet, A. S. H. V. E. Transactions, Vol. 40, 1934, p. 387.

Presented at the Semi-Annual Meeting of the American Society of Heating and Ventilating Engineers, June, 1936, Buck Hill Falls, Pa., by F. C. Houghten.

A complete description of the location, orientation and surroundings of the Grant Building, and the locations in the building of each of the rooms studied, as well as a detailed description of the arrangement of the test apparatus and methods used in collecting the data, are included in the earlier report,[1] and will be only briefly reviewed here.

The Grant Building is a 40-story, modern office building of the set-back skyscraper type, built in 1926 and located in downtown Pittsburgh. The location in the building and the exposure of the eight rooms studied are given in Table 1. Each room was heated between approximately 8:00 a. m. and 10:00 p. m. by electrical resistance heating units placed below the steam heating units in the convector cabinets under the windows, so that the convected heat was supplied to the rooms through the same grille and with approximately the same air temperature, velocity and direction as occurred in the normal heating of the

TABLE 1. ROOM CHARACTERISTICS

Room No.	Exposure				Length Corridor to Window	Transmission Loss for Entire Exposed Wall and Glass Btu/Hour/Deg F Temp. Diff. Based on the Guide 1935
	Direction	Width Ft	Window Area Sq Ft	Wall Area Sq Ft		
705	N. E.	8.48	40.75	42.62	9.75	64.0
707	N. W.	9.08	38.24	51.26	11.19	63.6
726	S. W.[a]	10.28	40.68	60.21	23.83	69.6
1826	S. W.	9.25	40.75	50.59	15.07	66.5
2703	N. E.	10.04	40.87	57.87	20.05	69.2
2706	N. E. & N. W.	10.33	79.66	123.14	10.33	139.5
2715	N. W.	9.46	37.91	55.09	11.74	64.8
2718	S. W.	10.42	40.62	61.81	20.04	70.4

[a] South 39° 30′ West.

building with steam. The electrical input was indicated by integrating watt-hour meters, read at frequent intervals to an accuracy which gave the heat input to within plus or minus 17 Btu. Approximately two-thirds of the required electrical input was hand controlled and on continuously. The remaining required heat input was controlled by an *on* and *off* thermostat. Each room was at all times controlled to the same average temperature as the surrounding space in the building, including the rooms above and below. This average was within plus or minus one degree of the actual temperature in any of the surrounding space. Between approximately 10:00 p. m. and 8:00 a. m., or from the close of one test to the beginning of the next, the electrical connections were broken and the normal steam supply and control of the building was used.

Thermocouples, read by a precision potentiometer, and mercury thermometers gave the temperature (hereafter called the control temperature) of the air at the breathing line in the center of the room where the thermostat was located, the temperature at various points between the floor and the ceiling, the temperature of the floor, ceiling and partition surfaces, the inside glass surface temperature, and the temperature of the air 3 ft outside of the window. Manometers

gave the pressure drop between the air within the room and that outside. Anemometers were used to give the vertical and the two horizontal components of the air movement 3 ft outside of the windows, while an indicating, integrating cup-type anemometer placed at the top of the building with the indicator on the 27th floor gave the miles of wind travel between observations. In most

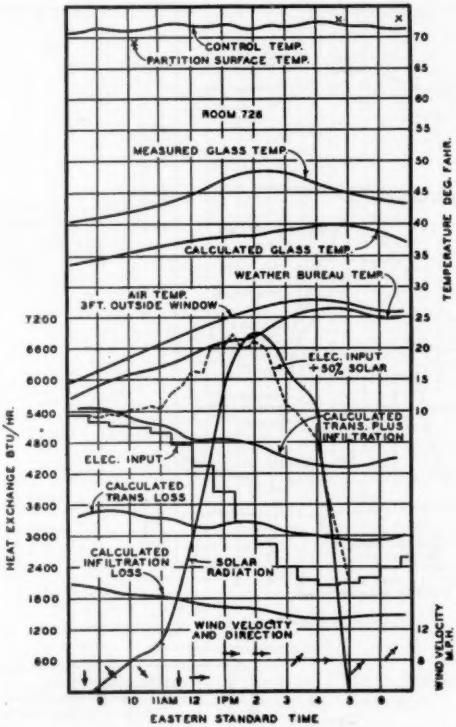


FIG. 1. LOG OF TEST ON FEBRUARY 6, 1934 IN ROOM 726, HAVING A SOUTHWESTERN EXPOSURE

of the tests the above observations were made at intervals of about 1½ to 2 hours. In a few of the tests these observations were made more frequently.

TEST RESULTS

Figs. 1 to 5, respectively, give the results of a test on February 6 in five rooms: 726, 1826, 2718, all having approximately equal exposures on Third Avenue or the southwestern side of the building, Room 2715 on the Grant Street or northwestern side of the building, and Room 2703 on Fourth Avenue

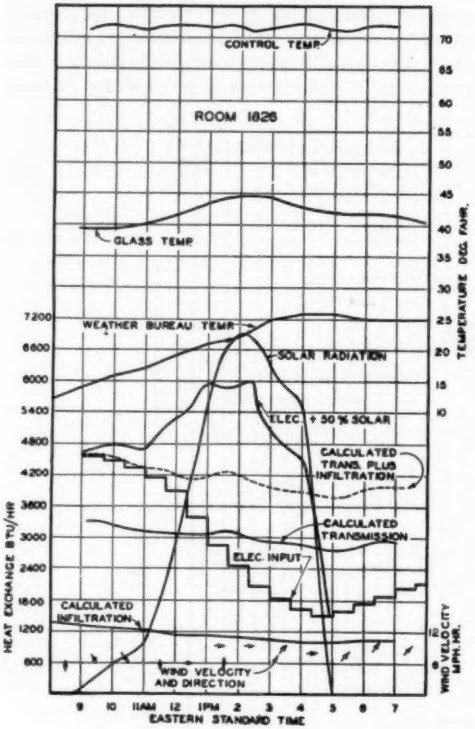


FIG. 2. LOG OF TEST ON FEBRUARY 6, 1934 IN ROOM 1826, HAVING A SOUTHWESTERN EXPOSURE

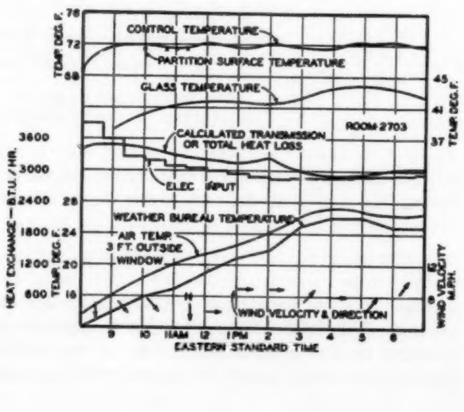


FIG. 3. LOG OF TEST ON FEBRUARY 6, 1934 IN ROOM 2703, HAVING A NORTHEASTERN EXPOSURE

or the northeastern side of the building. During this test all electrical input observations were made at 40 min intervals, and all other observations were made more frequently than was usually the case. The observed electrical input, the observed solar radiation incident upon the glass, and the sum of the electrical input plus 50 per cent of the solar radiation are plotted. The observed wind velocity and direction as given by the Weather Bureau are also plotted. The temperatures reported by the Weather Bureau, and in some

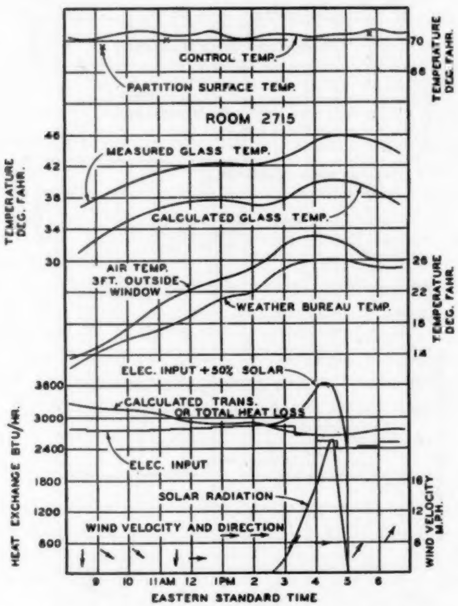


FIG. 4. LOG OF TEST ON FEBRUARY 6, 1934 IN ROOM 2715, HAVING A NORTHWESTERN EXPOSURE

instances those observed 3 ft outside of the window, the inside glass surface temperature and the room control temperature are also given. The calculated transmission heat loss, based upon the temperature 3 ft outside of the window, the calculated infiltration loss based upon the pressure drop through the window. and the sum of transmission and infiltration losses, or the total calculated heat loss from the room, are also plotted.

A log of weather conditions and the heat supplied to the several rooms, in Btu per average hour of test and per average degree temperature difference between the inside and outside air, for each test period are plotted in Fig. 6 for all tests from January 18 to March 29. In the upper section of the chart the wind velocity as reported by the Weather Bureau during the test period of each

day is plotted, and its prevailing direction during the same period is indicated by an arrow. The precipitation, measured in inches of water, is also given for the 24 hour period ending at 8:00 a. m. E. S. T., the following morning. The symbol *T* represents an undetermined trace of precipitation. The average outside air temperature during the period of each test and the mean 24 hour temperature as reported by the Weather Bureau are indicated by + and °, re-

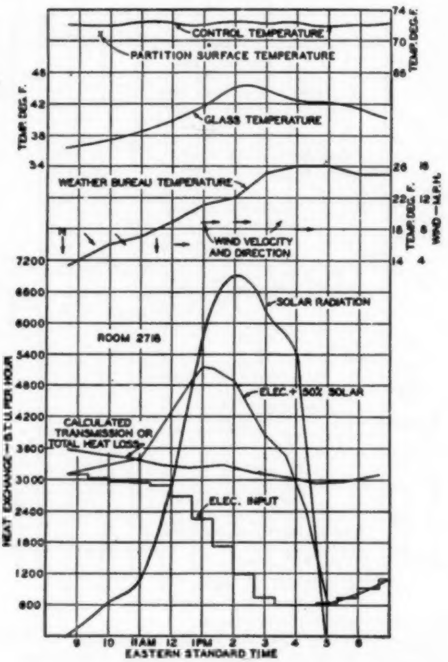


FIG. 5. LOG OF TEST ON FEBRUARY 6, 1934 IN ROOM 2718, HAVING A SOUTHWESTERN EXPOSURE

spectively, in the next lower section of the chart. The temperature range for each daily test period is indicated by a separate curve. In the eight lower sections of the chart the average heat requirement for the test period is given in Btu per hour per degree temperature difference between the temperature of each room and the outside Weather Bureau temperature. In each case the electrical heat input is indicated by an *x*, while for those rooms affected by solar radiation through the window, the sum of the electrical heat input and 50 per cent of the solar radiation incident against the glass of the windows is indicated by *o*.

The daily average heat loss in Btu per hour per average degree temperature difference between the inside temperature and the outside Weather Bureau recorded temperatures are plotted in Figs. 7 and 8 against the average wind velocity in miles per hour for each test period for the different tests in Rooms 705 and 2703, respectively. Both rooms are on the Fourth Avenue or northeastern side of the building and, therefore, subject to a negligible amount of solar radiation. The prevailing wind direction during each test is indicated.

The average hourly electrical heat input is plotted against outside temperature for all tests in Rooms 707, 726 and 2703 in Figs. 9, 10 and 11, respectively. Room 2703 on the northeastern side received practically no solar radiation, Room 707 on the northwestern side received very little, while Room 726 received considerable solar radiation. Points resulting from tests when there was no solar radiation are indicated as x in Fig. 10, for Room 726.

During the period of the study, the control of the steam supply to the building and observations on the rate of consumption were made available to the Research Laboratory staff. In Fig. 12 the daily steam consumption per degree day in the building is plotted against degree days, while in Fig. 13, the pounds of steam per hour per degree day are plotted against degree days. In Fig. 14 the average daily steam pressure in pounds per square inch maintained on the building side of the reducing valve is plotted against the daily mean temperature and the resulting degree days. Since the building is heated by a differential system, the steam pressure at this point is not very different from that existing throughout the system, and the steam temperature corresponding with these pressures as given in the chart is approximately the steam temperature throughout the system, or the temperature of the direct surfaces of the heating units.

Discussion of Results

The data plotted in Figs. 1 to 6 show marked variations in the hourly heat required per degree temperature difference to maintain the desired control temperatures in the several rooms, depending upon the elevation of the room above the street level, the solar radiation effect, and other factors.

Chimney Effect of the Building

Figs. 1, 2 and 5, giving complete logs of the test on February 6 in Rooms 726, 1826 and 2718, all having approximately the same exposure on the 7th, 18th and 27th floors, show calculated heat losses based upon transmission and infiltration, and heat inputs, including electrical input and electrical input plus 50 per cent of the estimated solar radiation incident on the glass of any room, which decreases in magnitude with elevation in the building. The average calculated heat requirement, the average electrical input and the average electrical input plus 50 per cent solar radiation for these rooms are, respectively 4747, 3677 and 4950 for Room 726; 4114, 2758 and 4154 for Room 1826; and 3215, 1784 and 3174 for Room 2718.

The average values for all tests in the several rooms given in Table 2 also indicate that the calculated heat requirements, electrical inputs, and electrical inputs plus one-half of the solar radiation decrease in magnitude with elevation in the building. This is particularly true when rooms of the same exposure,

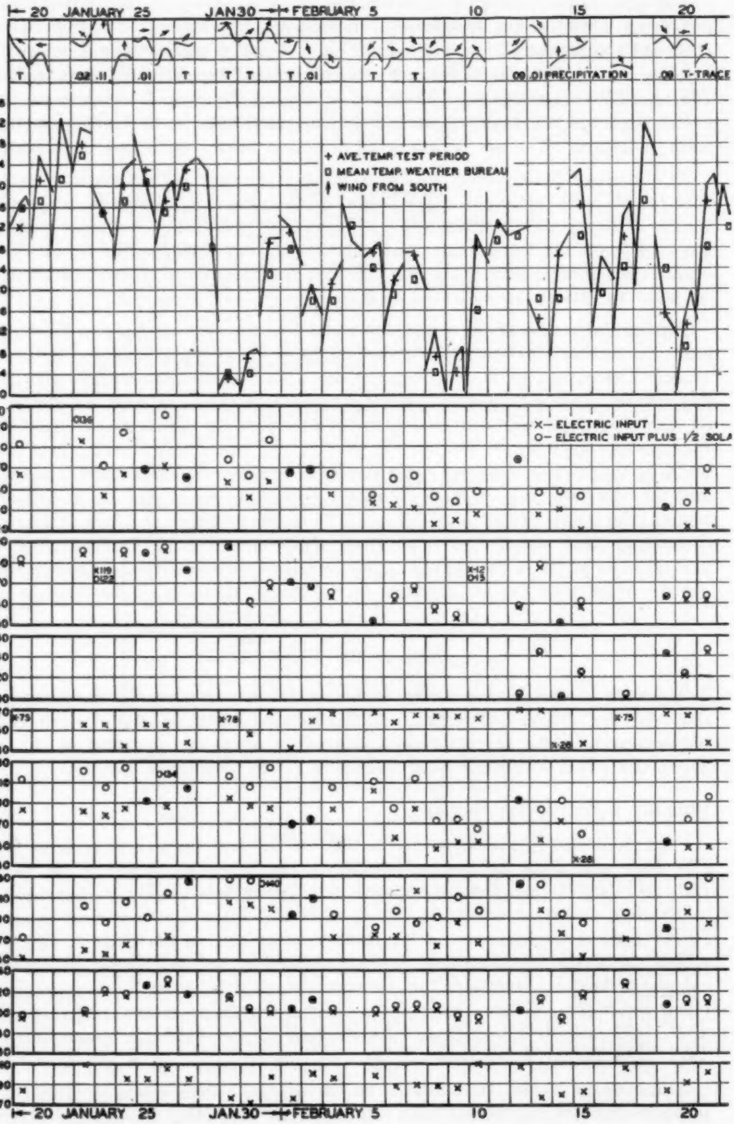
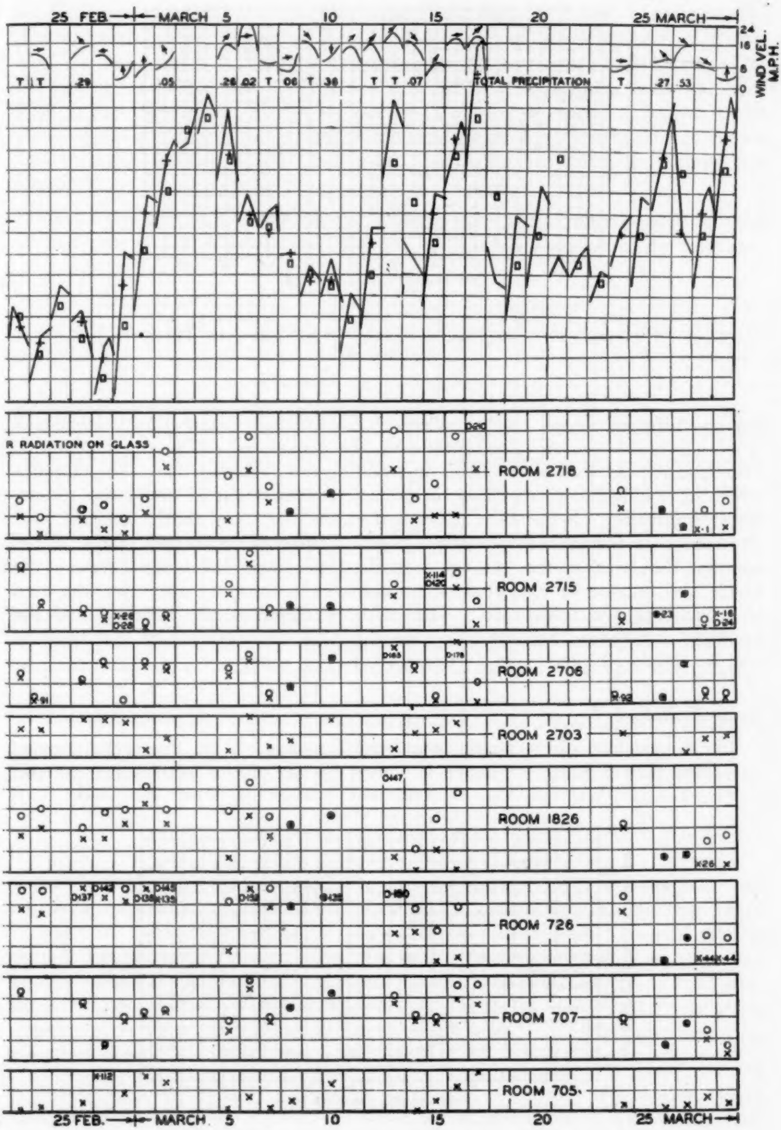


FIG. 6. LOG OF WEATHER CONDITIONS AND DAILY HEAT REQUIREMENTS IN BTU DIFFERENCE BETWEEN THE INSIDE AND OUTSIDE, DURING EACH TEST



PER AVERAGE HOUR OF EACH TEST PERIOD PER AVERAGE DEGREE TEMPERATURE PERIOD, FOR THE EIGHT ROOMS FROM JUNE 19 TO MARCH 29

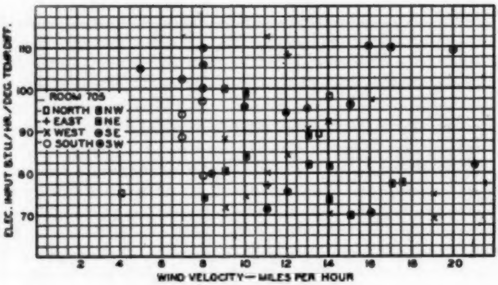


FIG. 7. RELATION BETWEEN WIND VELOCITY AND DIRECTION AND ELECTRIC HEAT INPUT FOR ALL TESTS IN ROOM 705

but on different floors, are considered. To a lesser extent this is also true when all rooms on the different floors are considered. These variations in heat requirement with elevation in the building are obviously due to the chimney effect of the building, resulting in excessive infiltration in the lower floors, lesser infiltration on the 18th floor, and exfiltration only in the average case for rooms on the 27th floor. This effect is reflected in the calculated values for the reason that the average measured pressure difference between the inside and outside of the window as observed in the different tests was used in calculating the infiltration. The chimney effect of the building and the resulting infiltration and its effect on the heat requirements of the building will be discussed further in a future laboratory report.

Solar Radiation

The tremendous effect which solar radiation has on the heat requirements of rooms with a sunny exposure is shown by the plotted results for the test

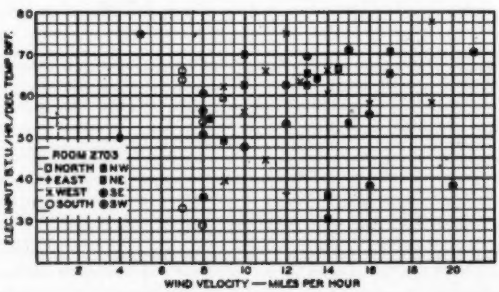


FIG. 8. RELATION BETWEEN WIND VELOCITY AND DIRECTION AND ELECTRIC HEAT INPUT FOR ALL TESTS IN ROOM 2703

on February 6 in Rooms 726, 1826 and 2718, Figs. 1, 2 and 5, respectively. The exposure of these rooms faces Third Avenue or the southwest. On this day, the sun's rays were first intercepted by the windows of these rooms at 8:42 a. m. and left them at 5:00 p. m., reaching a maximum value at about 2:00 p. m. The curves give the intensity of the solar radiation impinging on all the glass surface of these windows in Btu per hour. Since the three rooms had the same exposure and glass area, the curves in the three figures are identical.

Obviously, all of the solar energy striking the glass is not transmitted, and of that part which is transmitted through the glass and absorbed by the floor,

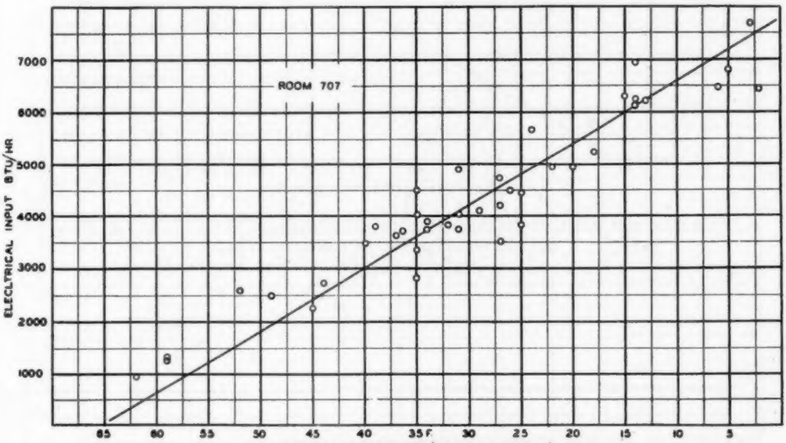


FIG. 9. RELATION BETWEEN OUTSIDE TEMPERATURE AND ELECTRIC HEAT INPUT FOR ALL TESTS IN ROOM 707

furniture and other surfaces, only a part is immediately given back to the air in the room. In another laboratory report [2] dealing with the cooling effect in some of these same rooms, it was found that during the summer cooling period approximately 50 per cent of the solar radiation impinging on the glass became effective in the cooling load during the cooling period from 7:00 a.m. to 4:00 p.m. E.S.T. For this reason, 50 per cent of the solar radiation impinging on the glass is added to the electrical input in establishing the curve, *electrical input plus 50 per cent solar radiation.*

It will be observed that for Room 2703, Fig. 3, which received no solar radiation during the test period, the electrical input correlates well with either

[2] Cooling Requirements of Single Rooms in a Modern Office Building, F. C. Houghten, Carl Gutberlet and Albert J. Wahl, A. S. H. V. E. TRANSACTIONS, Vol. 41, 1935.

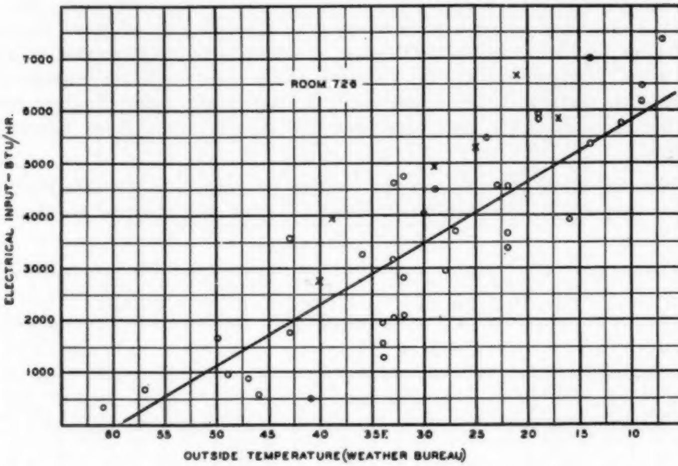


FIG. 10. RELATION BETWEEN OUTSIDE TEMPERATURE AND ELECTRIC HEAT INPUT FOR ALL TESTS IN ROOM 726

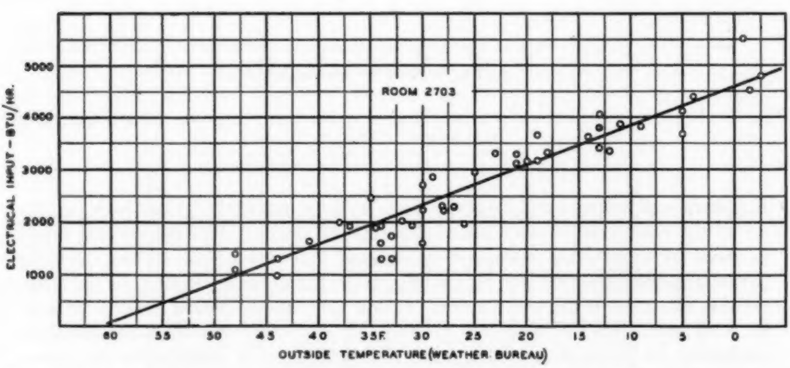


FIG. 11. RELATION BETWEEN OUTSIDE TEMPERATURE AND ELECTRIC HEAT INPUT FOR ALL TESTS IN ROOM 2703

the variation in outside temperature, or the calculated heat loss as plotted. For the other rooms which received solar radiation, the electrical input is considerably affected by the heat gain from solar radiation. However, the reduction in electrical input does not follow closely the increase in solar radiation, but lags considerably behind. Thus, for Rooms 726, 1826 and 2718, while the maximum rate of solar radiation impinging on the glass took place at 2:00 p.m., the minimum rate of electrical input resulting therefrom took place at times ranging from 3:00 to 5:00 p.m. for the different rooms. The curves for these rooms giving the electrical input plus 50 per cent of the solar radiation

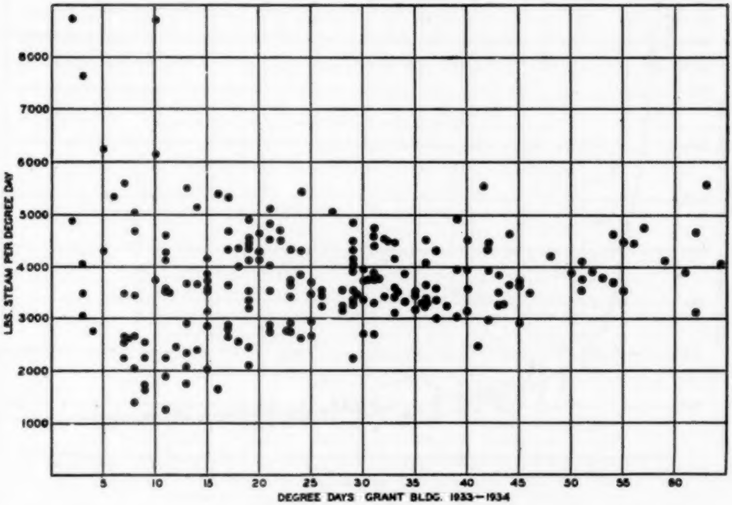


FIG. 12. RELATION BETWEEN DEGREE DAYS AND POUNDS OF STEAM CONSUMED PER DEGREE DAY IN THE GRANT BUILDING DURING THE 1933-34 HEATING SEASON

impinging against the glass, when compared with the curves for calculated heat loss, show that the combined heat input is excessive during the time when the solar radiation is effective and too small after the solar radiation ceases to be effective, showing that a considerable part of the solar radiation entering the room was stored as heat in the floor and furniture, to be given back to the air in the room at a later time.

The curves in Fig. 4 show less solar radiation effect in Room 2715 than in Rooms 726, 1826 and 2718. The time when the sun became effective in this room was later, the period of effectiveness was shorter, and the maximum intensity of the radiation was less. However, a similar lag between the time of the radiation and its effect on the heating load is shown.

It is apparent from the curves in Figs. 1 to 5, that solar radiation has a con-

siderable effect on the heat requirements of rooms affected, that this effect is much less than the total rate of radiation impinging against the window, and that a considerable lag exists between the time of maximum solar radiation against the window and its effect on the heat load of the room. The effect of solar radiation on the heating requirements of rooms having sunny exposures is further made apparent by the average hourly heat requirement of the several rooms for all tests plotted in Fig. 6. Rooms 705 and 2703, facing Fourth Avenue or the northeast, which received only a negligible amount of solar radiation

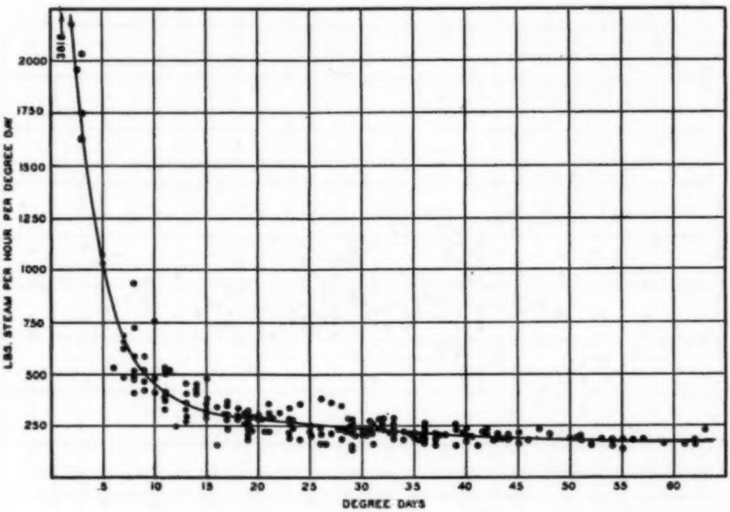


FIG. 13. RELATION BETWEEN DEGREE DAYS AND POUNDS OF STEAM CONSUMED PER HOUR PER DEGREE DAY DURING THE 1933-34 HEATING SEASON

in the early morning, show much less variation in the heat requirement than rooms receiving solar radiation.

It is apparent that on days during which solar radiation is a factor, electrical input is considerably less. However, the sum of electrical input plus 50 per cent of the solar radiation gives a variable value frequently considerably in excess of the electrical input for the same room on days when there is no solar radiation, indicating that great difficulty would be involved in properly evaluating the effect of solar radiation on the heat requirement of a given room on a given day. Obviously, solar radiation cannot be taken into account in estimating the maximum heating requirement of a given room or building, since the advantage can only be depended upon when the sun is shining. The sun effect is, however, a considerable factor in reducing the total heat required in such rooms during the heating season.

Variation in Heating Load with Outside Temperature

The electrical input required to maintain the desired temperature in Rooms 707, 726 and 2703 is plotted against outside Weather Bureau temperature in Figs. 9, 10 and 11. The individual test points plot more closely to the average curve for Room 2703, receiving no solar radiation, and spread out most for Room 726, Fig. 10, receiving the most solar radiation, where test points for days when there was no solar radiation are plotted as x. These points indicate a separate curve for such days, requiring a greater heat demand per degree temperature difference between the outside and inside, as should be expected, since the effect of solar radiation is to reduce the heat requirement by variable amounts, depending on the intensity and continuity of sunshine. The curve for all tests in Room 707, where solar radiation had a small effect, and a curve for Room 726, when there was no solar radiation, intersect the outside temperature

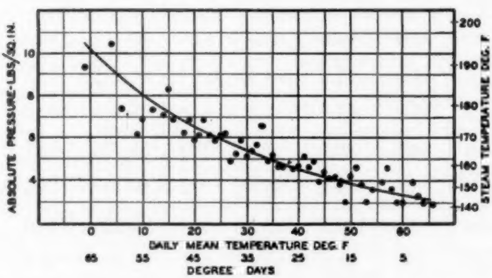


Fig. 14. Relation Between the Daily Mean Temperature and the Resulting Degree Days and the Average Daily Steam Pressure in Pounds Absolute Required in the Heating System of the Grant Building During the 1933-34 Heating Season

axis between 64 and 65 F outside temperature, indicating that for these conditions no heat is required with approximately 65 F outside temperature. Room 2703 on the 27th floor, which received no solar radiation, gives a curve intersecting the outside temperature axis at approximately 60 F. This difference appears to be due to the chimney effect of the building. On the 7th floor there was always a pressure difference from the outside to the inside, accompanied by a high infiltration rate which increased with lower outside temperatures, while on the 27th floor there was usually a pressure drop from the inside to the outside, resulting in exfiltration, causing no heat loss. All test points for Room 726, irrespective of solar radiation, give a curve intersecting the outside Weather Bureau temperature at about 60 F, while the lower limits of the points for this room would give a curve intersecting the outside temperature at about 50 F, showing that this room, based on an average of all tests irrespective of solar radiation, required heat when the outside temperature was about 60 F, and for most sunny days it did not require heat when the outside temperature

TABLE 2. RELATION BETWEEN THE CALCULATED AND OBSERVED HEAT REQUIREMENTS FOR THE SEVERAL ROOMS STUDIED IN THE GRANT BUILDING

A	B	C	D	E	F	G	H	I	J
Room No.	Calculated Trans. Btu/Hour/Deg F Temp. Diff.[a]	Calculated Infiltration Btu/Hour/Deg F Temp. Diff.[b]	Calculated Transmission and Infiltration Btu/Hour/Deg F Temp. Diff.	Meas. Elec. Input Ave. All Tests Btu/Hour/Deg F Temp. Diff.[c]	Meas. Elec. Input Calc. Trans. and Infil. (E/D) × 100 = Per Cent	Range Elec. Input All Tests Btu/Hour/Deg F Temp. Diff.	Elec. Input Plus 50 Per Cent Solar Rad. Ave. All Tests Btu/Hour/Deg F Temp. Diff.[c]	Meas. Elec. Input Plus 50 Per Cent Solar Radiation Cal. Trans. and Infiltration H/D × 100 = Per Cent	Range Elec. Input Plus 50 Per Cent Solar Radiation Btu/Hour/Deg F Temp. Diff.
705	64.0	32.0	96.0	88.0	91.7	70–112	88.0	91.7	70–112
707	63.6	34.0	97.6	104.3	106.9	65–128	107.5	110.1	65–134
726	69.6	34.2	103.8	87.8	84.6	44–135	106.5	102.6	54–180
1826	66.5	23.2	89.7	67.6	75.4	26–105	88.7	98.9	45–147
2703	69.2	0	69.2	55.0	79.5	28–78	55.0	79.5	28–78
2706	139.5	0	139.5	122.2	87.6	91–160	126.0	90.3	93–178
2715	64.8	0	64.8	59.6	92.0	12–119	63.3	97.6	15–122
2718	70.4	0	70.4	41.3	58.7	1–98	64.5	91.6	18–210
Ave. 726 1826 2718	...	...	...	...	79.2	...	...	97.7	...
Ave. 705 2703	...	...	...	...	85.6	...	...	85.6	...
Ave. 707 2715	...	...	...	...	99.4	...	...	103.8	...
7th Floor Ave.	...	...	...	...	94.4	...	...	101.4	...
27th Floor Ave.	...	...	...	...	79.4	...	...	89.7	...
Ave. All Rooms	...	...	...	...	84.5	...	...	95.3	...

[a] Col. B—Based on THE GUIDE 1935 heat transmission coefficients and 50 F temperature difference.
[b] Col. C—Based on the average observed pressure difference through the window for all tests, the observed relation between the pressure difference and air flow, and a 50 F temperature difference.
[c] Col. E and H—Average of all test values from Fig. 6.

reached 50 F. The data plotted in Figs. 9, 10 and 11 show the usually accepted straight line relationship between heat requirement and temperature difference.

Accuracy with Which Heat Requirement May Be Estimated

Further inspection of the heating requirements for the various tests and for the several rooms plotted in Fig. 6 shows variation which cannot be accounted for by solar radiation, elevation in the building, wind velocity, wind direction, or precipitation. As examples, the individual heat requirement points for Rooms 705 and 2703, where solar radiation was not a factor, show a range from 70 to 112 and from 28 to 78, respectively. It was shown previously that these variations could not be accounted for by wind velocity, direction, or precipitation. They appear, therefore, to be variations resulting from other uncontrolled factors in the study.

Since an attempt was made, insofar as possible, to take into account all variable factors, this variation would seem to be uncontrolled. In other words, it would indicate that taking all factors of exposure, types of construction, infiltration, etc., into account, the heat requirement of a given room still varies considerably from day to day. Any effect which variation in wind may have apparently comes within these limits and could not be evaluated from the findings of the study. In observing the heat requirement values for the different rooms, plotted in Fig. 6, there appears an apparent variation with time. Thus, Rooms 1826, 2715 and 2718 seem to show a higher heat requirement for the same temperature difference during the first few weeks of the study. No reason for this variation with time is apparent. It might possibly be accounted for by variations in moisture content of the building structure, or other factors beyond the attempted control of the study. The heat requirement, including the electrical input and 50 per cent of the solar radiation, expressed as a percentage of the calculated heat requirement in column I, Table 2, shows unaccountable variations for certain rooms. Room 707 shows a high percentage, while Room 2703 shows an unusually low value, which was consistent for all tests, indicating a low heating requirement. In part, this difference may be due to high infiltration in the case of 707, which is in the lower part of the building, and no infiltration in the case of 2703. This effect has, however, been taken into account in calculating the infiltration loss in column C.

Relation Between the Steam Required in the Entire Building and Outside Weather Conditions

In Fig. 12 the steam required per degree day is plotted against degree days for the 1933-1934 heating season. It is observed that less variation in steam consumption exists for colder weather. When the number of degree days per day is less than 20, a wide variation in the points is apparent. However, when the same points are divided by the number of hours of steam supply during any individual day and again plotted against degree days per day in Fig. 13, a very definite relationship is shown, giving a fairly constant rate of approximately 200 lb of steam per hour per degree day for days of 50 or more degree days, with an increasing rate of hourly steam consumption per degree day for milder weather. This further indicates the commonly accepted fact that heating requirement is more efficiently supplied in cold weather than in mild weather, and that the lack of efficiency becomes pronounced as the outside temperature approaches 65 F.

The data plotted in Fig. 14, giving the average daily steam pressure in pounds absolute per square inch required for varying daily mean temperatures and degree days are of interest. The differential vacuum system in the Grant Building requires a pressure difference between the steam on the building side of the reducing valve and that maintained by the vacuum pump of approximately 1 in. mercury column. Therefore, the pressure of the steam and the resulting steam temperature given in Fig. 14 indicate fairly accurately the mean heating unit surface temperature throughout the building and its relation to outside weather conditions.

SUMMARY AND CONCLUSIONS

1. This report gives data on the electrical input required to heat eight rooms located on three floors and having three different exposures in the Grant Building, Pittsburgh, to approximately 70 F.

2. The study briefly indicates a considerable reduction in heating requirement of the rooms on upper floors below that required for rooms on lower floors due to the chimney effect of the building. This subject will be discussed more thoroughly in another report from the Laboratory.

3. It is shown that the sun effect on the heating requirement of rooms subject to solar radiation is considerable and varies from no effect on cloudy days to a very large effect on days of more intense radiation. There is considerable lag between the time of maximum solar radiation effect and the resulting reduction in the heating load for any particular room. The percentage of solar radiation impinging against the glass of the window in a given room, which is effective in heating that room during the usual heating period of the day, varies considerably, but averages about 50 per cent.

4. The relation between heating requirement and temperature difference between the inside and outside air is indicated to be a straight line relationship, no heat being required when the outside temperature is above 65 F for the lower floors of the building, and at a slightly lower temperature for rooms in the upper part of the building where infiltration is not a factor. Solar radiation reduces the outside temperature at which heating is required on sunny days.

5. For best condition of control in rooms where solar radiation was not a factor during the test period, considerable variation in the heat requirement per degree temperature difference was observed which could not be accounted for by wind velocity or direction, or by precipitation.

DISCUSSION

M. C. BEMAN (WRITTEN): This paper is in the nature of a progress report by our Research laboratory which shows the progress of an investigation of the actual conditions existing in regard to solar radiation, chimney effects, etc., as well as a check of calculated heat losses in a building of this type. The first part of the investigation describing the methods pursued and general arrangements for the test were presented before the Society at the 1934 meeting and are published in the 1934 TRANSACTIONS.

In the previous paper, the method of determining the wind velocities and temperature at a distance of 3 ft from the building to establish outside weather con-

ditions and the comparison of these results with Pittsburgh Weather Bureau reports have been described. The work-up until the present time is clearly shown in the chart arrangement adopted. There will be further records from the following heating seasons which will give additional detail and permit more complete comparison of data in another paper which will be forthcoming at a later meeting.

As our buildings have increased in height a consideration called chimney effect, that upward draft through the building due to its height, has entered into our necessary calculations. In spite of cut-offs or partitions at each floor the necessary clearances around elevator and stair doors have to be such that an upward draft is established through the building which materially affects infiltration. This is definitely indicated in the log so far made of the Grant Building conditions. It is hoped that time and money will permit of securing sufficient data in regard to this phase of the study to enable the laboratory to establish the probable limits of the chimney effect for at least the type of interior and window construction used in the building and that approximate limits may be sufficiently established to provide more accurate data than now available to give the designer of the heating requirement the proper information for his work.

The laboratory is to be highly commended for the excellent progress made in this study and it is hoped it will be possible for our Research Committee to initiate further projects of this kind.

E. K. CAMPBELL (WRITTEN): Until such time as we can place greater dependence on the tightness of construction and control of leakage either inward or outward, refinements greater than we now have in calculating heat loads are of little value because of the unknown factor of air leakage. After a building is a year or two old an air leakage safety factor should be applied which upsets all the refinements of calculations that may be developed if the plant is to take care of its actual load under extreme conditions.

Competition and the desire to obtain a contract, coupled in some cases with ignorance of the recommended rules, lead frequently to under-calculation of heat loss, and the installation of unsatisfactory heating plants. It is, therefore, my opinion that the Society should not emphasize so much the refinements of calculating heat loss, as it should emphasize the desirability of delivering the required heat capacity under extreme conditions.

After considerable discussion in this paper, it was pointed out that the sun effect apparently has no bearing on the calculated heat load and should not be taken into account in determining the capacity of the installed plant.

In the paper reference is made to a general conclusion that a plant is inefficient when lightly loaded. I wish to call attention to the fact that inefficiency under such conditions applies to steam generating plants almost exclusively. It does not apply to either a hot water boiler or to a warm air furnace heating system. In the steam plant there is a definite low limit and below that temperature the plant does not seem to function properly. That is not true in either the hot water or the furnace plant. Hence the rule of inefficiency under light load in my judgment applies only to the steam generating plant.

The problem of overheating in mild weather is a problem in control and should not enter into the calculation of the capacity of the heating plant. If overheating is particularly objectionable, then that means that a system should be used which will best lend itself to the prevention of overheating and yet maintain ample capacity for the extreme load.

W. H. DRISCOLL: From the standpoint of a research study, this paper develops information of considerable interest, but I doubt if the findings of this research effort

would prompt any designing engineer to alter his method of calculation as to pipe sizes and radiation requirements.

I happen to have considerable familiarity with the building in which this research study took place, inasmuch as my company erected it, and not only did I spend a great deal of time on the job during the course of construction but our office made a careful check of the heating calculations and the pipe sizes. In this connection, we used the data contained in THE GUIDE, as we always do under such circumstances, with little or no question as to its accuracy. While it may be true, as pointed out by previous speakers, that there are some circumstances that require the exercise of personal judgment in the application of THE GUIDE data, I do not consider that as any reflection on the reasonable accuracy of the information it contains. In the practical application of this information, we are confronted with the necessity of using materials of commercial sizes. While it is not beyond the bounds of possibility that improvements in materials, equipment and devices used in heating plants may lead to greater refinements in design, the fact is that unless and until radiation can be made and sold in slices as thin as the ham in a cafeteria sandwich, and unless and until someone manufactures pipe so flexible that it may be reduced to fractional diameters by stretching it to greater lengths, we will always be confronted with the practical necessity of compromising between our calculated requirements and the nearest obtainable size of materials. Even if this were accomplished, there are many other factors that may have a possible effect on the heating requirements of a building which the designing engineer may not have the slightest control over. He is therefore compelled to use his personal judgment as to what factor of safety to allow in such cases.

In this particular building, for instance, the type of window was changed from that originally specified, and on which the consulting engineer would naturally have based his calculations, to a unique and unusual type on which information as to infiltration was not available. Furthermore, while the heating plant was being installed, the type of radiation was changed from a cast iron to a copper convector type.

Any attempt therefore at undue refinement in calculations as to radiation requirements would merely cause needless effort and the element of personal judgment had to be, of necessity, an important factor.

The real value of the paper, however, tends towards a consideration of possible improvement in operating methods to effect the utmost economy. This has always offered a fruitful field of investigation and I do not think we have reached the limit of possibility in that direction. This paper should be exceedingly helpful to those who are wrestling with that problem.

From the time that Andrew G. Paul brought out his vacuum system of heating, and more particularly in the last decade or so, rapid progress has been made in reducing the steam requirements for buildings, and especially large buildings. Much attention is being given to the subject by building operators, who are trying to solve the problem in a practical way, and by manufacturers and inventors, who are developing devices intended to accomplish such results. Not all of these devices, however, are practical nor economically desirable, but that is aside from the point.

A greater knowledge of what the minimum heating requirements may be under different conditions of wind, sun, and weather is of great importance but I doubt if it would be wise to make any substantial change in THE GUIDE data on this subject which for the designing engineer, it seems to me, is pretty reliable information.

No. 1046

APPLICATION FACTORS WHICH GOVERN THE SELECTION OF REFRIGERATING EQUIPMENT FOR AIR CONDITIONING SERVICE

By J. R. Hertzler* (*MEMBER*), York, Pa.

COMPRESSION type refrigeration equipment using dichlorodifluoromethane as the refrigerant is doubtless used in at least 90 per cent of all air conditioning installations made at the present time, for capacities up to 100 tons. There are no standards available which define the flexibility of operation required, the amount of condensing surface economically justified, the type of condenser and means of final heat rejection from the air conditioning system and the method of cooling and dehumidifying the air to be supplied to the conditioned space. It will be the purpose of this discussion to analyze the actual conditions of operation which a refrigeration system should be designed to meet when installed for human comfort air conditioning and to establish a basis for the proper balancing and selection of compressor, condenser and evaporator for usual conditions of operation encountered.

Refrigeration Load in Human Comfort Applications

Previous papers have described the cooling load factor in air conditioning, in which the refrigeration load was divided into its component parts:

1. Heat transmission through walls.
2. Sun effect.
3. Outside air load.
4. People.
5. Electrical load.

The first three factors, transmission, sun effect and outside air, will vary from zero to maximum, during the summer cooling season of four months. The internal loads, consisting of lights, people and any other source of internal heat such as fan motors, cooking equipment, escalator motors, or other equipment generating heat or moisture, are likewise variable but do not vary to the degree stated for the loads dependent upon the outside conditions.

The A.S.H.V.E. Guide 1936, Chapter 3, Table 2, specifies indoor tem-

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peratures recommended in summer, for various outdoor temperatures, which conditions are applicable for exposure of less than 3 hours. This gives a graduated scale of indoor temperatures to be produced for varying outdoor temperatures, arranged so that the dew point is held constant in the conditioned space, during summer operation, for dry bulbs ranging from 80 F with 95 F outside, down to 72 F inside for a 70 F outdoor temperature. It has been the experience of operators of human comfort air conditioning installations, where the systems have been installed for their promotional value, to produce a differential in temperature between the outdoor and the indoor temperature. Therefore any analysis of load factor which is based on a constant indoor temperature, such as the design temperature for the maximum outdoor dry-bulb, is not a true consideration of the load factor in accordance with recommendations of the Society and is further not a true load factor in accordance with the usual method of operation adopted by owners of air conditioning plants. From a record of hourly outdoor dry-bulb temperatures, tabulated in a previous paper,[1] the transmission load for a department store in New York City would pertain 486 hours out of a possible 1220 hours operation, provided the indoor dry-bulb design condition, in this case 80 F, were maintained throughout the summer operating period. On the basis of information obtained in this previous paper,[2] the load factor for transmission would be 12.9 per cent. if the indoor dry-bulb temperature were maintained constant at the design condition of 80 F throughout the summer. In accordance with THE GUIDE recommendations, the indoor wet-bulb temperature would also be dropped as the dry-bulb is reduced for lower outdoor temperatures. Any consideration of load factor which is made with the assumption that the indoor wet-bulb temperature will be constant throughout the summer operating season, is not in accordance with actual practice. Similarly, if the indoor wet-bulb temperature were maintained constant at 67 F, which is the indoor design wet-bulb temperature used for a maximum outdoor wet-bulb temperature of 75 F, the refrigeration system would need to operate only 548 hours out of a total of 1220 hours summer operation of the equipment, and the load factor due to heat from outside air would be 25.3 per cent.

In the following examples, it is assumed that air will be supplied to the conditioned spaces at a constant dew point temperature of approximately 54 F to produce an indoor dew point temperature of 57 F, in accordance with Table 2 of Chapter 3 of THE GUIDE. On this basis, refrigeration will be required to cool the outside air, whenever the outdoor wet-bulb temperature is above 54 F. It is further assumed that only outside air would be passed through the dehumidifier when the outdoor wet-bulb temperature is lower than the indoor wet-bulb temperature. From an analysis of United States Weather Bureau records of New York City for the years 1925 to 1931, the accompanying Table 1 indicates the number of hours from 8 a. m. to 5 p. m. daily, when wet-bulb temperatures between the limits tabulated occur. The average refrigeration load for the outside air and the ton hours of refrigeration per season for each of the groups of outside wet-bulb temperatures are also tabulated.

Dividing the operating season of the refrigeration equipment into five periods for outside wet-bulb temperatures as tabulated, that is, from 79 to 75 F, from

[1] What Is the Cooling Load Factor in Air Conditioning? by John Everetts, Jr., A. S. H. V. E. TRANSACTIONS, Vol. 40, 1934.

[2] Loc. Cit. See Note 1.

74 to 70 F, etc., the refrigeration load in ton hours per year required for cooling outside air would be 57,365 ton hours, out of a possible total of 114,500 ton hours for 1145 hours total operation, which results in a load factor of 50 per cent as indicated. This method of estimating load requirements due to outside air generally produces a result which is higher than the actual due to the fact that the final wet-bulb temperature calculated leaving the equipment is lower than the indoor wet-bulb temperature. However, the basis used is one which is believed to more nearly approach the actual than any form of analysis based on maintaining the inside wet-bulb temperature at the design condition for the assumed outdoor maximum. Another example will indicate a comparison using the actual indoor wet-bulb temperature rather than the apparatus temperature as the base for determining load factor due to heat added by outside air.

Table 1. Load Variations in Cooling Outside Air to 54 F Wet-Bulb

Outside Wet-Bulb Temp.	Hours	Ave. Load in Tons	Ton Hours
79F–75F	105	100.0	10,500
74 –70	275	74.5	20,500
69 –65	330	50.6	16,700
64 –60	275	29.1	8,000
59 –55	160	10.4	1,665
Total.........	1145 hours	....	57,365

Maximum Load at 77 F wet-bulb = 100 tons.

$$\text{Load Factor Outside Air} = \frac{57{,}365 \text{ ton hours}}{1145 \text{ hours} \times 100 \text{ tons}} = 50 \text{ per cent.}$$

The method of obtaining the overall load factor of an office building or department store internal load, which constitutes principally the electrical and people load, assuming population densities and lighting loads on a unit of square feet of floor area of the conditioned space, is shown in Fig. 1. Knowing the electric lighting load plus internal electric load in watts per square foot, on which the load factor is 100 per cent, and further knowing the square feet of floor area per person in the conditioned space, as well as an approximate load factor on the average number of people as compared to the maximum number of people for a given space, the overall load factor on light and people or the overall internal load factor may be determined directly from the curve. For example, with 2 watts of electric load per square foot, 40 sq ft of floor space per person and 25 per cent load factor for people, the overall internal load factor obtained directly from the curve would be 55 per cent. The curve indicates the load factors of three different factors for people loads of 25, 50 and 75 per cent. For other load factors on population, it is necessary to interpolate between the given load factors and the given electrical load curve.

Knowing the division of internal load, the load on a given job due to outside air and heat transmission, and the load factor on the internal load, it is possible from Fig. 2 to obtain directly the overall load factor for a particular installa-

tion. Thus with a 55 per cent load factor for an internal load, as determined from the previous example and a division of 40 per cent outside air and transmission load and 60 per cent internal load, the overall load factor for the plant under consideration would be 53 per cent. This value is obtained by reading vertically upward on the 40 per cent outside air and transmission load line to the 55 per cent load factor on internal load line, intersecting at 53 per cent overall load factor. In the department store example analyzed in a previous paper [3] the internal load on the department store was 80.5 per cent of the

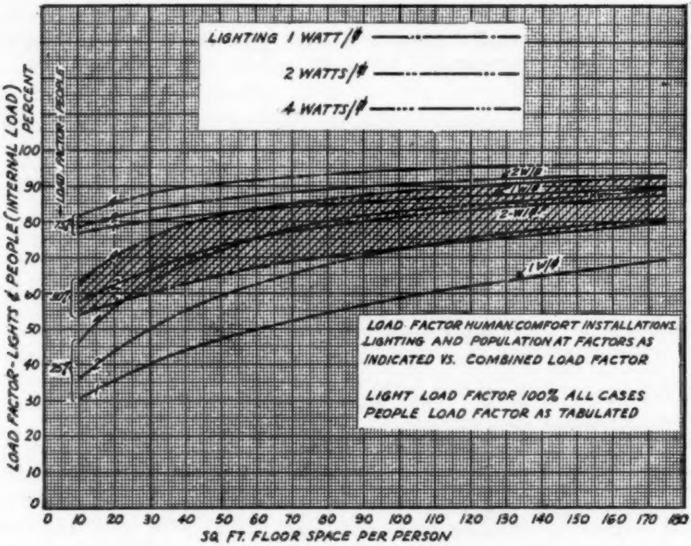

FIG. 1. INTERNAL LOAD FACTOR CURVES FOR VARIOUS FLOOR SPACES, OCCUPANCIES, AND LIGHT INTENSITIES

total and the outside air, transmission and sun effect were 19.5 per cent of the total.

It is proposed that the assumption be made that the transmission load and sun effect load vary approximately in the same proportion that the outside air load as measured by the wet-bulb temperature, assuming a constant dew point of 54 F leaving the air dehumidifier. For the condition of operation, as previously explained, the outside air load, transmission load and sun effect would have an overall seasonal load factor of approximately 50 per cent for most human comfort applications which may be extended to include eastern and central United States, excluding only the southern and far western portions of this country. On the basis of the above assumption, the load factor on human comfort air conditioning jobs approaches 50 per cent as the proportion of internal load to total load decreases and falls between 50 per cent and the

[3] Loc. Cit. See Note 1.

load factor on the internal load, for conditions of operation where the load factor on the internal load of a given installation is in excess of 50 per cent.

This analysis and the comparison with supplementary curves may be used only for buildings cooled during the period between 8 a. m. and 5 p. m. It is necessary to separate loads individually and analyze the individual conditions for theaters or restaurants, when the operating hours are different from 8 a. m. to 5 p. m. daily.

TYPES OF REFRIGERATING EQUIPMENT

Condensing Units

In the small sizes practically all manufacturers offer a line of standardized

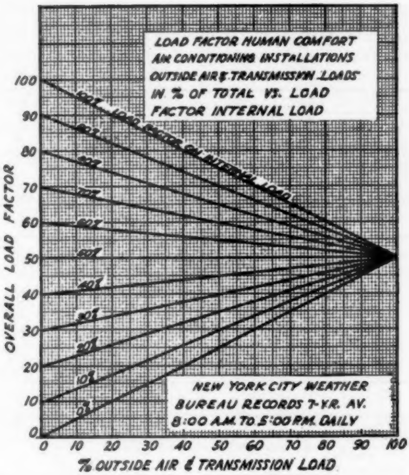

FIG. 2. OVERALL LOAD FACTOR CURVES FOR VARIOUS OUTSIDE AIR, TRANSMISSION AND INTERNAL LOAD FACTORS

self-contained automatic dichlorodifluoromethane condensing units, including an electrically driven compressor with a condenser of the shell and coil, double pipe, or shell and tube type, mounted in or on the compressor subbase. These machines are driven by electric motors at speeds in accordance with load requirements and the condensing surface utilized in the refrigerant condenser is proportioned so as to be applicable to a wide range of operating conditions. Operation of condensing units at evaporator temperatures used in human comfort air conditioning applications impose the greatest load on the refrigerant condensers. The refrigeration effect of the condensing unit is increased for increases in evaporator temperature, thus allowing a smaller amount of condensing surface per ton of refrigeration effect for air conditioning than would be the case should the same condensing unit be applied to cool a low temperature meat box maintained, for example, at 38 F.

Large Compressors

Above 40 hp, dichlorodifluoromethane compressors may be furnished with various types of drives, including electric motor, either direct connected to large synchronous motor driven compressors, or v-belt drive. It is likewise possible to operate large compressors by direct connection to diesel engines, gas engines or steam engines of the reciprocating type. Where operating economy dictates the use of low pressure or high pressure steam turbines, this type of drive can be applied to a reciprocating type compressor, by means of a combination speed reduction gear and belt drive assembly.

Because of the fact that refrigeration equipment is not uniformly rated at definite conditions as to evaporator and condensing temperature, it is difficult to rate a machine for a specific capacity, unless the actual operating conditions at which this capacity is obtainable are stated. It is desirable that the *American Society of Refrigerating Engineers* establish a method of testing and a method of rating dichlorodifluoromethane compressors at given standard air condition-

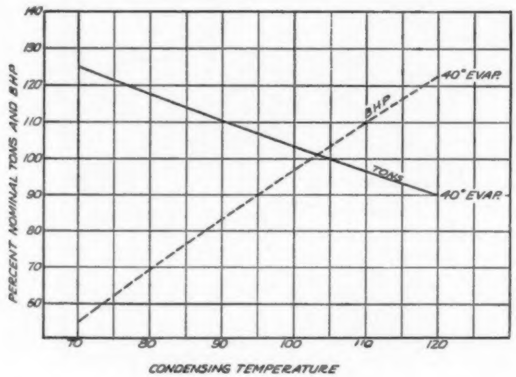

FIG. 3. CAPACITY VARIATIONS OF COMPRESSORS OPERATING AT A FIXED SPEED AND CONSTANT GAS VOLUME

ing evaporator and condenser temperatures, so that some of the confusion now prevalent may be eliminated. At the present time a proposed Standard Method of Rating and Testing Mechanical Condensing Units has been formulated for small units, but as yet no published ratings are available. The variation in capacity in tons of refrigeration is indicated in Fig. 3 for a given dichlorodifluoromethane machine, operated at a fixed speed and handling a fixed volume of gas. For condensing temperatures of 70 to 120 F and for a 40 F evaporator temperature, the capacity of this compressor varies from 125 to 90 per cent nominal tons. Throughout the same range of condensing temperature operation the brake horsepower varies from 55 to 123 per cent bhp nominal. In the lower portion of Fig. 4 tons of refrigeration and brake horsepower at a constant condenser temperature of 100 F for varying evaporator temperatures from 20 to 50 F are shown. It is therefore apparent that the compressor operated by the same motor at 20 F would have a capacity of 66 per cent nominal tons of refrigeration when requiring 96 per cent nominal brake

horsepower and would have a capacity of 127.5 per cent nominal tons of refrigeration and require only 93 per cent nominal brake horsepower at 50 F evaporator temperature. The absolute necessity of fixing the operating conditions for a given compressor is indicated by reference to these curves. The upper portion of Fig. 4 shows the cost of power in cents per ton hour for a

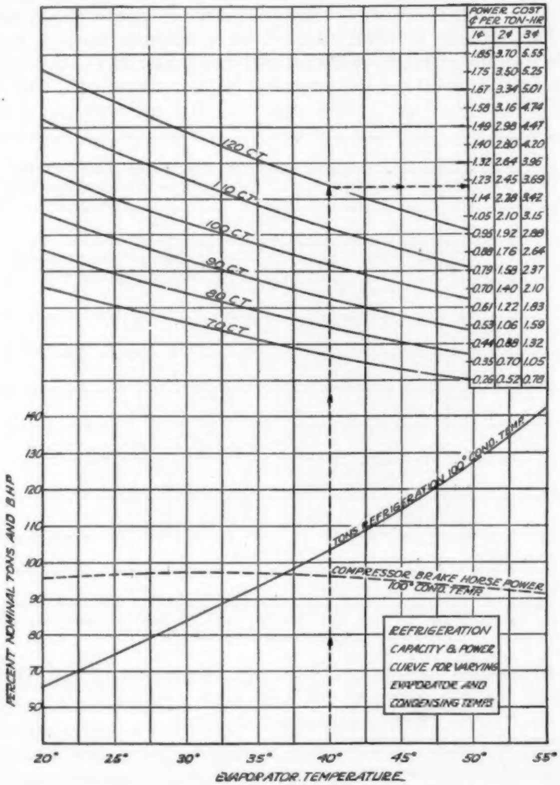

FIG. 4. REFRIGERATION CAPACITY AND POWER CURVES FOR VARYING EVAPORATOR AND CONDENSING TEMPERATURES

wide range of evaporator and condensing temperature conditions at which the compressor is operated. These power costs are tabulated for a unit price of 1, 2 or 3 cents power cost per kilowatt hour. A further application of these curves will be illustrated in a later example.

Condensers

In refrigeration systems, other than the unit type, condensers of various types, sizes and capacities are frequently used with large compressors as previ-

ously described. In the past large installations of condensers of the shell and tube type have been used where a supply of condensing water was available, either from city mains, deep wells or cooling tower systems. These condensers were generally designed so that they could be readily retubed in the field and were therefore installed using bare pipe surface of steel or non-ferrous materials. While it is true that the condensing temperature obtainable with various types of condensers is not always directly in proportion to the amount of condensing surface installed in a given condenser, Fig. 5 indicates an approximate operating condition obtainable under normal conditions of operation for varying quantities of condensing surface installed. The difference

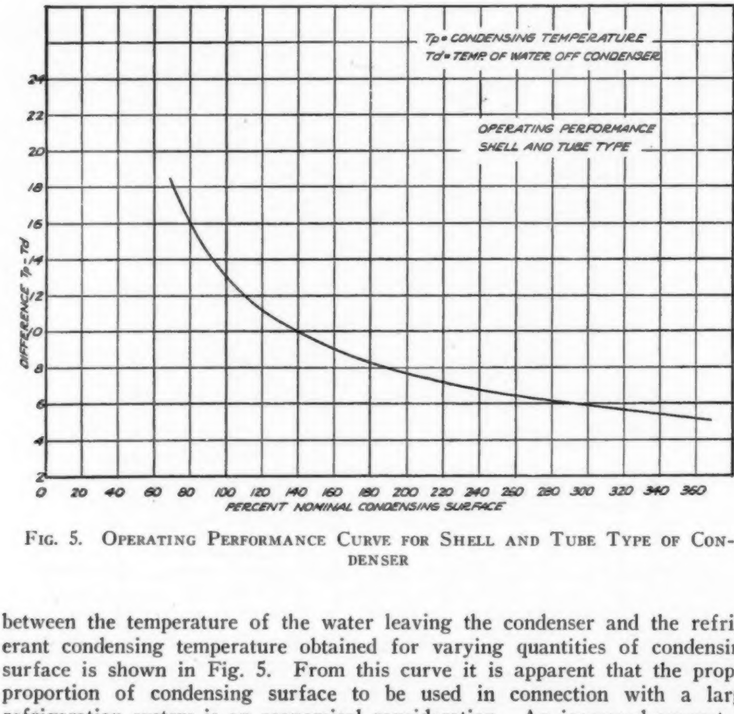

FIG. 5. OPERATING PERFORMANCE CURVE FOR SHELL AND TUBE TYPE OF CONDENSER

between the temperature of the water leaving the condenser and the refrigerant condensing temperature obtained for varying quantities of condensing surface is shown in Fig. 5. From this curve it is apparent that the proper proportion of condensing surface to be used in connection with a large refrigeration system is an economical consideration. An increased amount of condensing surface will provide a reduced operating cost for a dichlorodifluoromethane refrigeration system and will involve an increased first cost to the consumer. The limit to which condensing surface should be added for improving operating economy is thus dependent upon the power saving which would be obtainable with a larger condenser. It is likewise apparent that the condensing surface furnished for a given compressor would vary in accordance with the cost of power on individual installations and would vary for cooling tower installations as compared to city water installations.

A basis for determining the economical limitations of design and cost are shown in Fig. 6 which indicates the difference between the final condensing water temperature leaving a cooling tower and the outside wet-bulb temperature entering the cooling tower, for various conditions of load. This curve is based upon the assumption that at no load, the water would leave the cooling tower 2 F above the entering wet-bulb temperature, whereas for full load the difference between the leaving water and entering wet-bulb temperature would be 8 F. Naturally these conditions vary in accordance with the tower design but they are approximate average values which may be used for this basis of comparison. For conditions of less than full load, the difference between leaving water and entering wet-bulb temperature varies in direct proportion

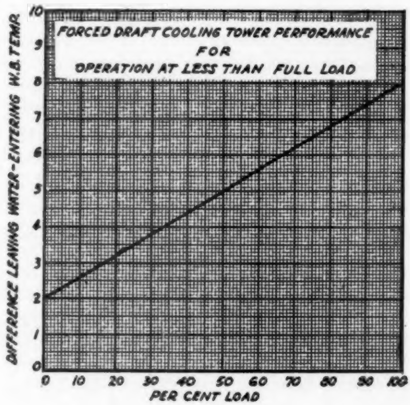

Fig. 6. Forced Draft Cooling Tower Performance Curve for Varying Load Capacities

to the load on the tower. The analysis of an individual problem of condenser selection for a cooling tower installation of 100 tons capacity, operated from 8 a. m. to 5 p. m. in the eastern territory, with a load factor of 60.5 per cent overall is graphically shown in the curves of Fig. 7. This installation at maximum load required 100 tons of refrigeration, of which 47.5 per cent of the total is due to outside air and transmission, and 52.5 per cent of the total is internal load on which there is an overall internal load factor of 70 per cent. In this example a nominal condenser rating of 10 sq ft of condensing surface per ton was used. From an analysis of these curves it will be noted that for power at 1 cent per kwhr, 8 sq ft of condensing surface per ton is most economical. For power at 2 cents per kwhr, 9 sq ft of condensing surface per ton provides the lowest annual fixed and operating cost for the condenser. For a power cost of 3 cents per kwhr, 10 sq ft of condensing surface per ton is most economical. For power at 4 cents per kwhr, condensing surface in the amount of 10 sq ft per ton may be justified on the basis of this analysis of fixed and operating costs on the condenser. The proportion of fixed charge to

operating cost may be noted by comparing the fixed charge curve shown at the bottom in dotted line with the total charge as indicated by full line on the cooling tower system.

For purpose of comparison, the total annual fixed and operating cost of a condensing system when utilizing city water with a final temperature leaving the condenser of 93 F throughout the year is shown for a 1 cent power cost. It will be noted that the power required by the refrigerating compressors is 30 per cent greater utilizing city water maintained at a constant discharge temperature of 93 F, than would be the case with a cooling tower system of the forced draft type utilizing outside air for cooling the condensing water in the system. This power saving for the compressor would be offset by the water pumping cost required for circulating the condenser water through the condenser and the cooling tower and would further be offset by the fan horsepower utilized by the fans for a cooling tower of the forced draft type. Since the condensing water cost was not included in this comparison, the economy of installing a cooling tower is not established by this analysis. It is generally true, however, that the power required for operating the refrigeration compressor in an air conditioning system is less for a cooling tower installation than would be the case with a city water system, in which the amount of city water used is controlled from the discharge temperature of the city water leaving the condenser.

The steps necessary in making a complete condenser cost analysis are indicated in Table 2.

For the five groups of wet-bulb temperatures it was necessary to establish the refrigeration load at the average wet-bulb temperatures for each group. The difference between the water leaving the tower and the wet-bulb temperature is shown in Fig. 8, which establishes the leaving water temperature off the tower. The difference between condensing temperature and temperature of the water leaving condenser may be obtained from Fig. 5. The cooling tower water range is in proportion to the refrigeration tonnage for the particular load condition with the maximum load imposing a range of 10 F on the quantity of condensing water circulated. This condition would be on the basis of 3 gpm per ton, which would increase the temperature through a 10 F range on ordinary air conditioning applications. By adding the difference in temperature values obtained from Fig. 5 plus the cooling tower water range to the water temperature leaving the cooling tower, the refrigerant condensing temperature is obtained. The evaporator temperature in each case is in accordance with the refrigeration load in the water cooler. For a full load of 100 tons, the temperature difference between the average chilled water, which is assumed to be 50 F, for producing a 54 F leaving air temperature at the dehumidifier is 10 F. That is, the amount of surface in the water cooler is proportioned so that with an evaporator temperature of 40 F and an average chilled water temperature of 50 F the cooler will have the required capacity of 100 tons. Then for each condition of load as tabulated, the temperature difference would be in proportion to the load on the cooler. The power cost in cents per ton hour is selected for the exact evaporator and the condenser temperatures obtained for each case in accordance with values given on Fig. 4 for power at 1 cent per kwhr. The ton hours of refrigeration load is the product of the tons refrigeration load for the various wet-bulb temperatures

under consideration and the number of hours at which these wet-bulb temperatures occur per year. The operating cost for each wet-bulb condition is the product of the ton hours and the cost of refrigeration power in cents per ton hour, as tabulated.

TABLE 2. COST ANALYSIS OF COOLING TOWER APPLICATION

Wet-Bulb Temp. Range	Tons Refrig.	Diff. Water Leav. Tower and Wet-Bulb Temp. (Fig. 6)	Temp. Water Leav. Cool Tower	Cool Tower Water Range for 300 gpm.	Tp-Td (Fig. 5) 80 Per Cent Nom. Sq Ft	Cond. Temp.	Evap. Temp. 40 F at 100 Tons	Cents /Ton Hour 1 Cent /Kwhr (Fig. 4)	Ton Hours	Operating Costs
79–75	84.25	7.1	84.10	8.4	13.7	106.2	41.6	0.88	8850	$ 77.80
74–70	72.15	6.5	78.30	7.2	11.9	97.4	42.8	0.73	19820	144.80
69–65	60.75	5.65	72.65	6.07	10.4	89.1	43.9	0.58	20000	116.00
64–60	50.55	5.05	67.05	5.06	9.1	81.2	44.9	0.46	13900	64.00
59–55	41.75	4.5	61.50	4.2	7.9	73.6	45.8	0.35	6680	23.40
										$426.00

Wet-Bulb Temp. Range	Water Leav. Cond.	Tp-Td 90 Per Cent Nom. Sq Ft	Cond. Temp.	Cents /Ton Hour	Operating Costs	Tp-Td 100 Per Cent Nom. Sq Ft	Cond. Temp.	Cents /Ton Hour	Operating Costs
79–75	92.5	12.25	104.8	0.87	$ 77.00	11.3	103.8	0.84	$ 74.30
74–70	85.5	10.8	96.3	0.70	139.00	10.1	95.6	0.68	135.00
69–65	78.7	9.5	88.2	0.57	114.00	8.8	87.5	0.56	112.00
64–60	72.1	8.3	80.4	0.44	61.20	7.7	79.8	0.44	61.20
59–55	65.7	7.3	73.0	0.34	22.70	6.7	72.4	0.34	22.70
					$413.90				$405.20

Wet-Bulb Temp. Range	Water Leav. Cond.	Tp-Td 120 Per Cent Nom. Sq Ft	Cond. Temp.	Cents /Ton Hour	Operating Costs	Tp-Td 150 Per Cent Nom. Sq Ft	Cond. Temp.	Cents /Ton Hour	Operating Costs
79–75	92.5	9.8	102.3	0.83	$ 73.50	8.3	100.8	0.80	$ 70.80
74–70	85.5	8.8	94.3	0.67	133.00	7.4	92.9	0.65	129.00
69–65	78.7	7.7	86.4	0.55	110.00	6.6	85.3	0.53	106.00
64–60	72.1	6.8	78.9	0.43	59.80	5.9	78.0	0.42	58.40
59–55	65.7	6.0	71.7	0.33	22.10	5.1	70.8	0.32	21.40
					$398.40				$385.60

Combined Cooling Tower Condensers

Where a forced draft tower is applicable or where it is desired to locate a cooling tower indoors, it is possible to build the refrigerant condensing coils in the cooling tower spray chamber which reduces the usual water pumping cost required in an ordinary forced draft cooling tower water system. The

fan horsepower would be comparable to a forced draft cooling tower of the usual design and condensing temperatures obtainable are dependent upon the amount of condensing surface installed in the combined cooling tower condenser per ton of refrigeration, and the final wet-bulb temperature of the air leaving the combined cooling tower condenser.

There are many variable factors affecting the design of a combined cooling tower condenser and a definite basis for rating this type apparatus is of even greater importance than is the case in other air conditioning equipment.

The capacity of a given cooling tower condenser varies with changes in each of the following operating conditions:

1. Evaporator temperature.
2. Condensing temperature required.
3. Entering air wet-bulb temperature.
4. Air volume supplied and final wet-bulb temperature of leaving air.

For example, a unit cooling tower condenser may be rated for a given capacity with an evaporator temperature of 30 F, a condensing temperature of 110 F, an entering air wet-bulb temperature of 75 F and a fixed quantity of air supplied for the final means of heat rejection from an air conditioned space.

Capacity changes due to operation at other than rated evaporator temperatures are small, with a 7.5 per cent increase in capacity for a 30 F rise in evaporator temperature to 60 F, and an 8.5 per cent decrease in capacity for a 30 F reduction in evaporator temperature to 0 F.

Rated capacities are related to the condensing temperatures to be produced in the condenser of a dichlorodifluoromethane refrigeration system. If an outside wet-bulb temperature of 75 F and a fixed quantity of air for a cooling tower condenser were required to produce a condensing temperature of 90 F instead of 110 F rated, its capacity would be 38.4 per cent of the nominal capacity at 110 F condensing temperature. Were the condensing temperature permitted to rise to 120 F instead of 110 F the nominal rated capacity of the apparatus would be increased 35.5 per cent all other factors remaining constant.

The variation in capacity of a combined cooling tower condenser with changes in the wet-bulb temperature of the entering air is shown in Fig. 8. If the wet-bulb temperature at which the equipment is rated were dropped from 75 to 65 F the capacity would be increased 17.5 per cent all other variables remaining constant and if raised from 75 to 85 F the capacity would be decreased 22 per cent.

Other factors governing the design of this type of equipment have a definite bearing on capacity such as the type of water distribution whether sprayed or of the Baudelot type, air velocities over condensing surface, depth of condensing surface in direction of air flow or actual type of condensing surface whether finned or plain pipe.

The ease of cleaning and maintenance with this piece of equipment is exceedingly important inasmuch as the source of air supply may be contaminated to a considerable degree.

It is of greater importance when making operating power comparisons to consider the condensing temperature rating of the cooling tower condenser rather than the actual fan and motor pump horsepower requirements. In

most cases the power requirements of the refrigeration compressor will be ten or more times that of the cooling tower condenser so that economies of operation may more readily be obtained by reducing condensing temperature than would be possible by refinements of fan and pumping arrangements. Due to the relatively greater importance of the heat of the liquid of dichlorodifluoromethane a reduction in condensing temperature will likewise cause an increased capacity of the refrigeration compressor in addition to a decreased

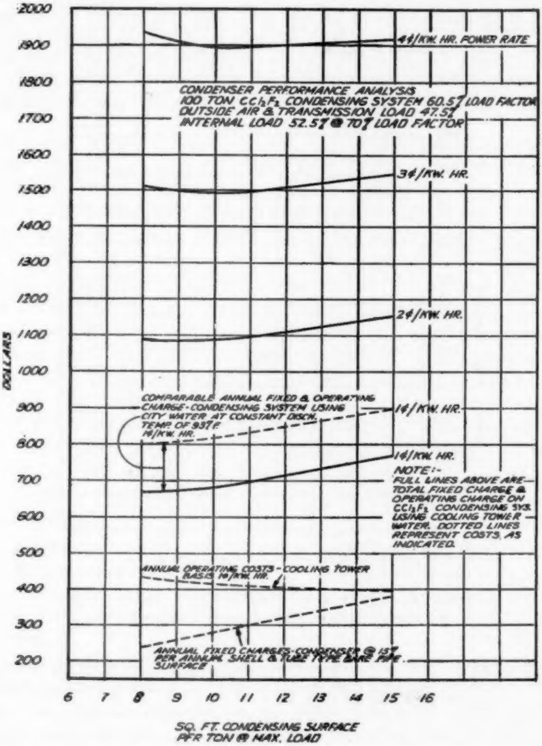

FIG. 7. ECONOMIC CONDENSER SELECTION CURVES FOR DIFFERENT SURFACE AREAS AND POWER COSTS

power input to the compressor motor. Therefore a reduction in condensing temperature is of greater importance than subcooling of the refrigerant because of reduced power requirements as well as increased capacity as the compressor displacement remains constant.

Evaporators

Evaporators may be furnished for dichlorodifluoromethane air conditioning applications of the direct expansion fin surface type. Fin surfaces are furnished

in many styles and types, such as helical, flat, corrugated, and compound fins, which are usually bonded to the pipe coils by means of a tinning operation. In most cases the thermostatic type of expansion valve is used for fin surface direct expansion air conditioning applications.

It is likewise possible, in the application of air cooling with direct expansion, to locate the cooling coils in the spray chamber of an air dehumidifier. This method of design obtains all of the advantages of a spray type dehumidifier, without enlarging the cross section of the equipment, provided the evaporator coils are properly designed. Because of the high heat transfer obtained with a water spray on the exterior surface of the direct expansion evaporator piping, fins are of no particular advantage, with the result that bare pipe coils have an installation advantage in the spray chamber of air dehumidifiers. The

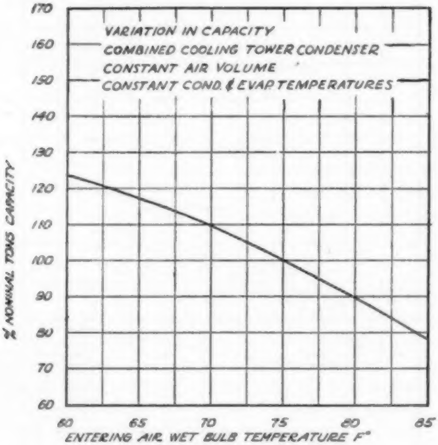

FIG. 8. CAPACITY VARIATION CURVE FOR COMBINED COOLING TOWER CONDENSER

flooded evaporator method of operation is preferable to thermostatic expansion, when the coils are located in the water spray of a dehumidifier, because of the smaller temperature difference used for economical compressor selection and operation. One of the principal advantages of a dehumidifier system is that the air leaves the apparatus in a saturated state, so that the relative humidity in the conditioned space is more closely controlled. Furthermore a duct system designed for the same inside temperature and humidity may be smaller in area with an air dehumidifier of the spray type than with a surface coil type installation. In the intermediate seasons, that is, spring and fall operation of the air conditioning equipment, the sprays of the air dehumidifier can be used to obtain evaporative cooling on days when the operation of the mechanical refrigeration system would be required with a surface cooling installation.

The indirect method of refrigeration may similarly be applied to both surface cooling and air dehumidifier spray type coolers, with the water cooled by a separate refrigeration plant and circulated to the air cooling surface or the air

dehumidifier. The majority of all human comfort air conditioning installations made to date in the large sizes have been designed with a water spray in preference to the surface cooling type of installation. However, surface cooling has become more popular in the past few years.

APPLICATION OF REFRIGERATION EQUIPMENT

From the previous discussion it is evident that a thorough analysis of the load factor is necessary before the proper selection of refrigeration equipment may be made for an air conditioning installation.

The minimum capacity at which a plant will be required to operate will determine the minimum number of refrigeration compressors necessary or will dictate the number and types of capacity reducing devices which will be installed for a given compressor as selected. Most large dichlorodifluoromethane reciprocating compressors are equipped with hand or automatically operated capacity reducing devices which will permit economical operation at partial loads down to approximately 50 per cent of the maximum capacity. If the load analysis indicates that a further reduction in capacity will be necessary for certain periods of operation this can be applied to a single compressor installation by furnishing speed reduction on the compressor motor down to 50 per cent of maximum speed, thus reducing the capacity of the compressor to 25 per cent of maximum by the combination of capacity reducing devices on the machine itself and reduced speed operation. Adjustable speed control is especially desirable on an installation where the driving motors operate on direct current because of the economical operation obtainable by means of field control of a direct current adjustable speed motor. On alternating current installations multispeed motors are applicable up to 150 hp and where direct connected synchronous motors are used it is desirable to increase the number of compressor units to obtain the minimum refrigeration capacity and flexibility of operation required.

Where refrigeration units are installed in multiple for a given installation it is generally advantageous to have the compressors installed so that all of the condensing surface and all of the evaporating surface can be utilized for all conditions of operation, which results in a direct operating economy as shown in Table 2.

It has been indicated that the kilowatt input per ton to an electrically driven dichlorodifluoromethane compression type refrigeration system is reduced for partial load operation provided the full condensing and evaporator surface is utilized at all loads. It was shown in a previous paper[4] that economy dictated the use of 9 sq ft of condensing surface per ton. When operated at ½ capacity, with the evaporator temperature held constant, the brake horsepower per ton of refrigeration was reduced 9 per cent. By raising the evaporator temperature 5 F at ½ load operation, with the condensing temperature held constant, the brake horsepower per ton would be decreased 9.5 per cent or combined, the saving by using the full condensing and evaporating surface for operation at ½ load would be 17.7 per cent in brake horsepower per ton. Since the compressor motor efficiency was 4 per cent lower at ½ speed than at full speed the full saving of 17.7 per cent was not obtained but the net saving was

[4] Operating Results of an Air Conditioning System Compared with Design Figures, by J. R. Hertzler. A.S.H.V.E. TRANSACTIONS, Vol. 40, 1934.

sufficient to justify the installation of a single cooler and condenser to be used with a variable compressor capacity. Operating costs may be reduced as much as 25 per cent by utilizing this arrangement of equipment selection.

It is therefore desirable to parallel condensers or evaporators, or to install more than one compressor to operate on a given condenser and water cooler. This method of application may be considered a standard arrangement of design because of the resultant reduced operating costs when using dichlorodifluoromethane equipment. The operating economy of a refrigeration system at partial load is far more important than the economy at full load because the refrigeration system operates at less than full load more than 90 per cent of the time.

CONCLUSIONS

As a basis for the selection of dichlorodifluoromethane refrigeration equipment for air conditioning service the following standards may be established.

1. The load factor and the actual loads at which the equipment will be required to operate determine the number of compressor units or the capacity regulating devices of the individual compressor units required.

2. The fixed charges on a particular refrigeration equipment should be calculated at 15 per cent of the initial cost per annum.

3. The cost of the driving energy and the first cost of the prime mover of the compressor generally determine the type of drive selected.

4. The amount of condensing surface justifiable is variable and dependent upon operating costs or the balance of first and operating cost which results in the lowest annual expense to the owner.

5. Use of combined cooling tower condensers may be justified on the basis of reduced annual fixed and operating charges or by other causes such as inadequate or improper condensing water supply or limited sewer capacity.

6. Evaporators are generally selected to provide the desired accuracy of control, flexibility of operation, ease of installation and the lowest seasonal fixed and operating charges.

7. As practically all systems must be operated at less than full load the greater part of the time, a method should be provided to operate the compressor economically at reduced capacity. Where flexibility of operation is provided by multiple compressor units, the evaporating and condensing surface installed with such units should be available for all conditions of load to provide the greatest operating economy.

DISCUSSION

L. L. LEWIS: There is one important point in this excellent piece of work which cannot be over-emphasized. Not only is air conditioning a seasonal load but also one having a very low load factor. Load factors for the refrigerating machine will rarely exceed 50 per cent. Therefore, too much study cannot be put upon examination of partial load characteristics of refrigerating machines.

It is refreshing to get away from what appears to be the mad scramble to select equipment which will carry the maximum load and which can be installed at minimum cost.

No matter what our practices may be today, they cannot prevail indefinitely if they are economically unsound. In thinking of economic unsoundness, we must not only include the engineer, who designs the plant, and the owner, but also the public utility which supplies the service required for its operation.

I recently had the good fortune to come into the possession of an integrating demand meter record of an air conditioning plant employing about 200 tons of refrigeration. Since these records were taken for the purpose of studying the load characteristics of the plant, it was possible to determine exactly what each of the several motors was doing. The plant conditioned the first floor and basement of a large department store and the records revealed a surprisingly low load factor of 53 per cent for 102 days of operation.

The electric utility which supplied current stated that, in this particular instance, they required a revenue of $24.00 per year per connected kilowatt in order to break even. An analysis of the record revealed that, while some of the smaller motors paid them as much as $35.00 per year per connected kilowatt, others dropped to as low as $5.70.

Studies were made of alternative designs of refrigerating plants and revealed the interesting fact that one plant might bring a loss on a gross business of some $16,000.00 per year, whereas another one would return a nice profit upon a gross business of less than $9,000.00 per year.

Air conditioning will continue to increase the maximum demands imposed upon generating plants. As this increases, the time will come when additional generating equipment and transmission facilities will have to be installed in order to meet it. Where this will come first must necessarily depend upon the local situation, and questions cannot be avoided if and when it becomes necessary to invest additional capital for business that may, at least, not be profitable because of the characteristics of its demand.

The foregoing applies directly to the air conditioning plant. Fortunately, when the load on the entire building is considered, the picture is changed somewhat because the load curve of a particular store has a valley during the summer months without an air conditioning load. It is, of course, evident that it is the entire building load that counts rather than any particular part of it.

In Memoriam

Names	Joined the Society	Died
Victor W. Cherven	1928	Oct. 1936
N. Loring Danforth	1919	Aug. 1936
Edward Glanz	1930	July 1936
Alfred L. Hadesty, Jr.	1921	Apr. 1936
Joseph H. Leonard	1931	Feb. 1936
Jay R. McColl	1916	Oct. 1936
Rowland J. Millar	1925	July 1936
Charles E. Monday	1920	Mar. 1936
Mathieu L. Sakouta	1924	Dec. 1936
Percival H. Seward	Charter Member	Nov. 1936

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